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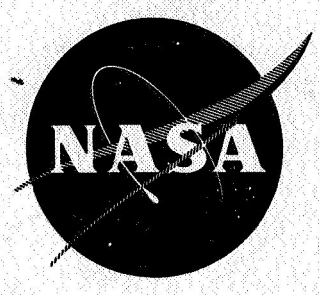

BEARING FATIGUE INVESTIGATION

by

E.N. Bamberger

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

Contract NAS 3-7261

FLIGHT PROPULSION DIVISION

GENERAL ELECTRIC

LYNN, MASSACHUSETTS/CINCINNATI, OHIO

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FINAL REPORT

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September 15, 1967

CONTRACT NAS 3-7261

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FOREWORD

The work reported herein was performed at the facilities of the Flight Propulsion Division, General Electric Company, Cincinnati, Ohio, under NASA Contract NAS 3-7261. The NASA Project Manager was Mr. D. Townsend and the NASA Research Advisor was Mr. E. V. Zaretsky,

The author would like to acknowledge the efforts of the following personnel who aided significantly in theperformance of the program. Mr. C. C. Moore under whose supervision the full scale bearing tests were performed and who was instrumental in the conception and design of the bearing testers. Mr. D. Smeaton who guided the construction of the testers and supporting facilities. Messrs. C. Johnson and R. Brooks who performed the actual bearing tests. Mr. D. Kroeger who performed the RC Rig testing and Messrs. J. Fausz and P. Schneider who performed respectively the lubricant analysis and the bearing weight and dimensional measurements.

BEARING FATIGUE INVESTIGATION

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ABSTRACT

A research program has been performed to establish the feasibility of operating large diameter rolling element bearings at 600F under conditions of load and speed approximating those experienced by main shaft bearings in jet engines. Two bearing materials (CVM M-50 and WB-49) and three advanced lubricants were investigated. The lubricants were: a) synthetic paraffinic oil, Mobil XRM-177F, b) polymeric perfluorinated fluid, DuPont PR-143, and c) inhibited, mixed isomeric 5-ring polyphenyl ether 5P4E, Monsanto MCS-354. Over 54,000 hours of bearing testing were accumulated. Additionally, a large number of bench type rolling contact fatigue tests were conducted, as well as extensive lubricant and metallurgical analysis, The data shows that the paraffinic oil will provide satisfactory hydrodynamic lubrication at 600F in an inerted atmosphere. The polyphenyl ether and the polymeric fluid are marginal in their lubricating capability at the 600F temperature. Bearing failures at 600F are similar to those observed at lower temperatures. CVM M-50 is suitable for bearing applications up to and including 600F.

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1.0 Summary and Conclusions

A research program has been conducted to establish the feasibility of operating large diameter rolling element bearings at 600F under conditions of load and speed approximating those experienced by main shaft bearings in jet engines. Two major bearing materials and three fluids were evaluated. Over 54,000 hours of bearing testing were accumulated, 27,000 of which were at 600F. Additionally, a considerable number of bench type (RC Rig) rolling contact fatigue tests were conducted, as well as extensive lubricant and metallurgical analyses. The major conclusions are:

- A synthetic paraffinic oil will provide satisfactory bearing operation at 600F under a nitrogen atmosphere.

- Using the synthetic paraffinic oil there appears to be no significant effect of temperature on bearing operation in the range 400F-600F.

- Bearing failures at 600F are similar to those experienced at lower temperatures, i.e., sub-surface initiated spalling fatigue.

- At 600F, with the synthetic paraffinic oil and M-50 bearings, bearing life is over 2 times computed life or 12 times catalog life.

- At 600F in an air atmosphere a modified polyphenyl ether does not provide satisfactory lubrication, resulting in considerable component wear and surface distress.

- At 600F, using M-50 bearings, a polymeric perfluorinated fluid generally provides adequate lubrication, although the adequacy of the lubricant film is not consistent.

- WB-49, an alternate high temperature bearing material, is significantly inferior to M-50 when compared at 600F and with the synthetic paraffinic oil as a lubricant.

- S-Monel retainers are highly satisfactory for 600F operation.

2.0 Introduction

The phenomenal growth of the aircraft industry in the last decade, spurred on primarily by the rapid advancement in propulsion systems, has placed unprecedented demands on materials and lubricants engineering. Traditionally, bearing operating temperatures were dictated by the thermal capability of the lubricating fluid. Bearing structural materials such as M-50, M-1, 440C, etc. had temperature capabilities exceeding available synthetic lubricants such as the diesters and polyesters. Within the last few years, however, a number of new fluids have been developed which show promise of a breakthrough in terms of increasing the usable temperature range of lubricating oils. Among these are the polyphenyl ethers, the synthetic oils and the polymeric perfluorinated fluids.

One of the critical problems foreseen in the operation of supersonic and hypersonic aircraft will be the high ambient temperature to which the engine and gearbox bearings will be exposed. These components are normally lubricated by oil which also acts as an effective heat transfer medium, passing much of the heat generated by the bearings to an available heat sink such as the fuel. With increased speeds, however, and particularly in the Mach 3+ regime, the heat transfer capacity decreases due to the inherently higher temperature of the fuel and greater demands for cooling capacity elsewhere. This, coupled with the increased heat generation by the bearings due to higher speeds and heavier loads, requires improved lubricants and to a lesser extent, better structural materials for the bearings,

The formulation of advanced thermally stable oils is, however, only the first step in this quest for bearing systems capable of higher temperatures, Design data relating the performance of these fluids and bearing materials at the elevated temperatures must be obtained. Relationships vital to bearing design and life predictions such as load/life and temperature/life need to be determined. Extrapolation of existing data simply is not good enough since totally new families of fluids are

being utilized, or the more common fluids are being operated at their maximum capacity and are thus in an area of essentially unpredictable performance.

The availability of these new lubricating fluids, along with initially encouraging basic engineering properties indicating their potential for high temperature bearing operation, prompted NASA to undertake a study of these fluids under conditions of load, speed and temperature approximating those seen in main shaft jet engine bearings.

General Electric, under contract to NASA–Lewis Research Center, has been performing a portion of these studies. The primary emphasis of the General Electric studies was the determination of bearing performance in the temperature range 400F-600F. The bearings used were 120 mm bore, split inner ring, angular contact bearings.

The program as originally conceived was designed to evaluate three new families of high temperature fluids in combination with two high temperature bearing steels. Based on the results of these studies, a load–life test series was then to be conducted with the best fluid–bearing material combination. This latter test sequence was subsequently changed to the determination of a life–temperature relationship. The reason for this modification will be explained in a later section of this report.

Supporting the full scale bearing tests were a number of complementary studies designed to aid in the analysis of the bearing test data. Included in these supporting tests were rolling contact fatigue studies (RC Rig), computer analysis of bearing kinematics and stresses, as well as lubricant and metallurgical evaluations.

3.0 Lubricants and Bearing Materials

The program was designed to provide an evaluation of the operating characteristics of three advanced lubricants, at elevated temperatures and under realistic operating conditions of load and speed. To provide the greatest chance of achieving this prime objective, the prime bearing material selected was CVM M-50, a Cr-Mo-V high speed tool steel, currently used by all major jet engine manufacturers in high temperature and/or critical bearing applications. In addition to this material, an alternate high temperature bearing material was included, in order to have a back-up in the event the M-50 was found to be unable to provide satisfactory operation at the proposed test temperatures of 600F. This alternate material was Crucible Steel's WB-49, a high alloy tool steel, specifically developed for high temperature bearing applications.

Specific details regarding the test fluids and bearing materials are presented in the following pages.

3.1 Lubricants

The three lubricants selected for evaluation were:

a. Synthetic paraffinic oil - Mobil XRM-177F
b. Polymeric perfluorinated fluid - DuPont PR-143
c. Inhibited, mixed isomeric 5-ring polyphenyl ether, 5P4E, - Monsanto MCS-354

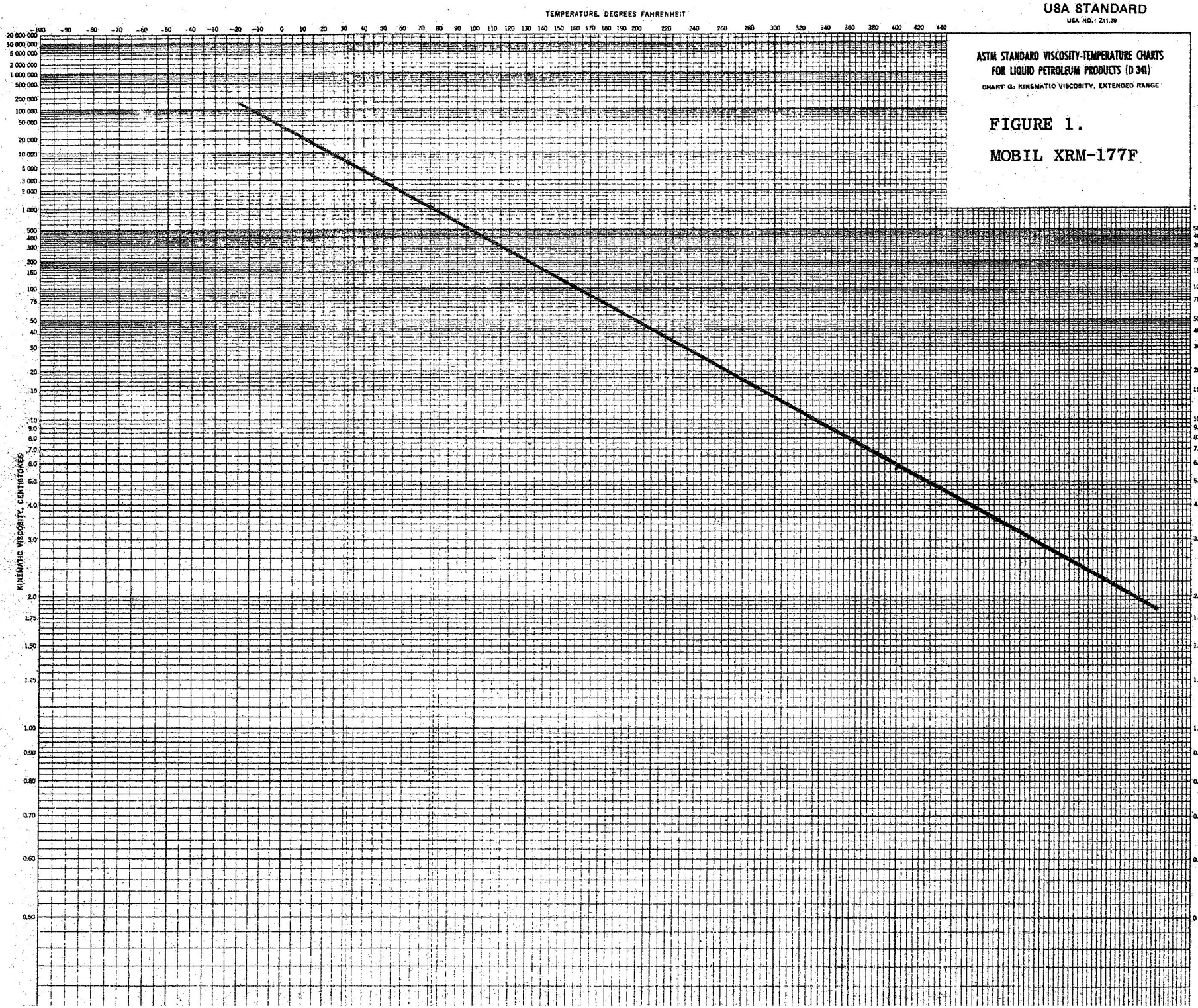
Mobil XRM-177F is a 100% synthetic paraffinic oil having excellent high temperature viscosity characteristics. As originally planned, the oil to be used in the General Electric work was to be Mobil XRM-109F, which is the same fluid minus an EP additive. However, before the GE testing was initiated, SKF reported[(1)]* that the EP additive was essential to achieve satisfactory life from this fluid. Consequently, the XRM-109F on hand was returned to Mobil Oil Company for modification into the XRM-177F. Physical property data furnished by Mobil Oil Company is presented in Table 1 and Figure 1.

* See Table of References.

Table 1

Properties of XRM-177F-2

Kinematic Viscosity (CS)	
0F -	37,497
100F -	443.3
210F -	39.72
400F -	5.83
Viscosity Index	124
Pour Point, °F	-35
Flash Point, °F	515
Fire Point, °F	600
Autogeneous Ignition, °F	805
Volatility (6.5 hrs. @ 500F)	14.2%
Specific Heat	(See Table 3)
Thermal Conductivity	(See Table 3)


USA STANDARD
USA NO.: Z11.39
ASTM STANDARD VISCOSITY-TEMPERATURE CHARTS
FOR LIQUID PETROLEUM PRODUCTS (D 341)
CHART G: KINEMATIC VISCOSITY, EXTENDED RANGE
FIGURE 1.
MOBIL XRM-177F
TEMPERATURE, DEGREES FAHRENHEIT
KINEMATIC VISCOSITY, CENTISTOKES

DuPont PR-143 is a polymeric perfluorinated high temperature fluid having high oxidative and thermal stability. Shortly after its development, the material was evaluated by General Electric under rolling contact conditions and was found to be capable of providing satisfactory bearing operation up to and including 600F.[2] This fluid is extensively documented in an Air Force report[3] and pertinent physical property data of the batch used in the General Electric Program is shown in Tables 2 and 3.

Monsanto MCS-354, the third fluid is an inhibited mixed isomeric 5-ring polyphenyl ether 5P4E, developed for use in high-temperature hydraulic, heat transfer and lubricant applications. The properties of this fluid have been extensively documented.[4] The pertinent physical properties for the batch of material used in the current program are shown in Tables 3 and 4.

3.2 Bearings and Bearing Materials

The test bearings used in this program are shown in Figure 2 (GE Dwg. #4012286-956). A detailed stress analysis of this bearing is presented in section 5.0 of this report, and the major critical dimensions, ball diameters, contact angles, etc. are shown in Figure 2.

The initial order of bearing components for this program was as follows:

	Component	Quantity	Material
1.	Outer Ring	158	CVM-M50
2.	Outer Ring	90	WB-49
3.	Inner Ring	158	CVM-M50
4.	Inner Ring	90	WB-49
5.	Retainer	130	S-Mone 1
6.	Retainer	120	M-1
7.	Balls	2686	CVM-M50
8.	Balls	1530	M-1

Table 2

Properties PR-143AC, Lot 5

Density @ 75°F, g./ml.	1.8798
Viscosity, cs.	
@ 0°F	32,062
@ 100°F	298.3
@ 210°F	29.81
@ 400°F	4.55
Viscosity Index (ASTM D-567)	123
Viscosity Index (ASTM D-2270)	146
ASTM Slope	0.562
Pour Point, °F	-30
Volatility, wt. per cent 6 1/2 hrs. @ 500°F	18.0
Oxidation - Corrosion Data @ 650°F. 72 hrs. with 5 liters dry air/hr: Metals, mg./cm.^{2} - day	
QQ-S-636	-3.15
301 SS	-2.74
M-50 Tool Steel	-2.67
440C SS	-3.74
Ti (8Mn)	-4.61
% Fluid Loss	
Cell	0
Overall	0
% Viscosity Change	
100°F	+8.0
210°F	+3.6

Table 3

Specific Heat and Thermal Conductivity

Of Three Test Fluids

		PR-143	MCS-354	XRM-177
1. Specific Heat[1]	100F	0.241	0.35	0.475
	200F	0.260	0.39	0.530
	300F	0.279	0.44	0.585
	400F	0.279	0.48	0.640
	500F	0.316	0.53	0.695
2. Thermal Conductivity[2]	100F	52.1	84.0	79.0
	200F	52.0	81.0	77.0
	300F	51.8	79.0	75.0
	400F	51.6	78.0	72.0
	500F	51.5	78.0	70.0

(1) Specific Beat - BTU/lb/^{0}F

(2) Thermal Conductivity - BTU/hr/(ft)^{2} (^{0}F/ft) x 10^{3}

Table 4

Properties of MCS-354

Viscosity (CS)

lOOF	358.0
210F	13.0
400F	2.1
500F	1.2
600F	0.85

Pour Point, °F	+40F
Vapor Pressure	0.76 mm @ 500F
Specific Gravity	1.200 @ 25/25C
Refractive Index	μ_D^{25} 1.6307
Flash Point	540F
Evaporation Loss 6 1/2 hrs. @ 500F @ 140 mm	8.5%
Fire Point, °F	660
Autogeneous Ignition Temp., °F	1135
Specific Heat	(See Table 3)
Thermal Conductivity	(See Table 3)

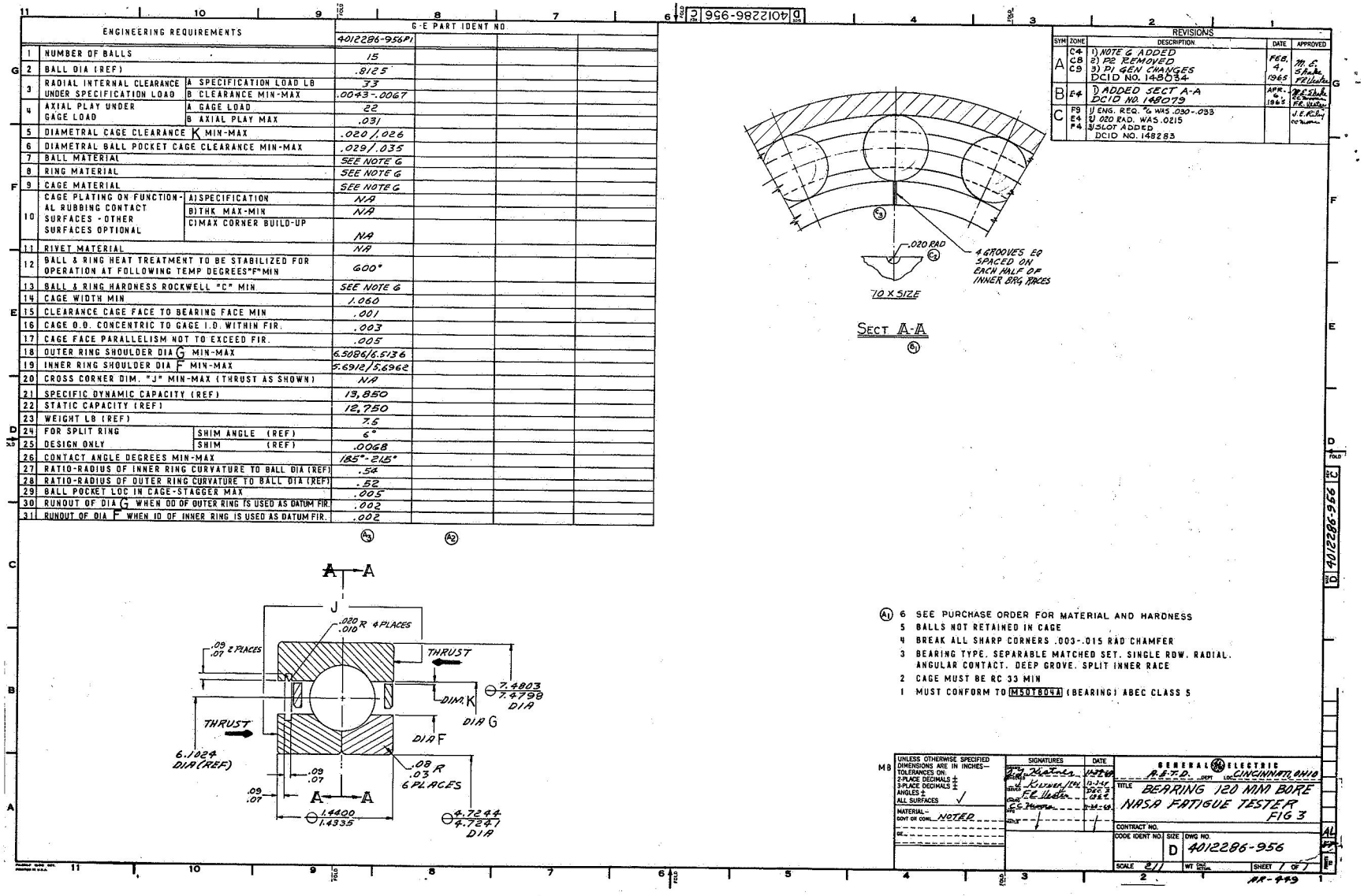


FIGURE 2 120 MM BORE ANGULAR CONTACT BEARING USED FOR HIGH TEMPERATURE LUBRICANT EVALUATION

The materials selection was made on the premise that the majority of the testipg would be performed with the M-50/S-Monel combination. In the event that the S-Monel cages proved marginal, a substitution to M-1 cages was to be made. Similarly, sufficient WB-49 components were procured in the event that the M-50 would prove unsuitable and/or marginal, for the 600F operation, Based on NASA experience,(5) it was also decided to outfit the WB-49 bearings with M-1 balls, as this latter material had provided satisfactory operation, especially when exposed to the PR-143 fluid,

The metallurgical aspect of the bearing procurement was a particularly stringent one, and was essential to insure the repeatability and credibility of the bearing test data. The most salient points of the metallurgical control were as follows:

1. All bearing components made from the same material were to be made from one heat of material.

2. Hardness requirements for the various components were as follows:

M-50 Rings	Rc 62-64
WB-49 Rings	Rc 64-66
M-50 Balls	Rc 62.5-63.5
M-1 Balls	Rc 64.5-65.5
S-Monel Retainers	Rc 33 ± 1
M-1 Retainers	Rc 55 ± 1

3. Raceway surface finish was not to exceed two r.m.s.

The above conditions were successfully met, with the exception of the M-50 components in Item 1 (above). In this case, insufficient material was available from a single CVM billet to produce all components.

Consequently, this requirement was relaxed to allow the M-50 balls to be made from separate CVM billets, although these were still from the same heat as the one used for the rings. (It should be noted that the interpretation of a material "heat" is a controversial subject and the author would like to clarify General Electric's position on this subject. There are essentially two "heats" as such in a consumable vacuum melting process. The first of these is the air melted master heat from which as many as six CVM electrodes can be obtained to produce individual CVM billets. The master heat can be as much as 15-20,000 lbs., whereas a CVM billet rarely exceeds 3,000 lbs, Despite very close control of the CVM process, sufficient variations can occur during the process, so that General Electric considers each CVM billet an individual heat, and thus not necessarily the same as a similar billet made from the same master material, but melted at a different time.) Table 5 presents the Chemical Analysis of the materials used in the current program, To control the hardness tolerances, specifically on the balls, two extra balls were made for each bearing specifically for hardness checks.

To illustrate the hardness control achieved, a summary of hardness measurements is presented in Tables 6 through 10,

Additionally, from each heat of material with the exception of those used for the retainers, RC Rig test bars were made and submitted to General Electric for evaluation. These tests and their results are discussed in another section of this report. To further assure the soundness of the bearing material, X-ray diffraction measurements were made to determine retained austenite contents. The following excerpt from a Marlin Rockwell Bearing Company letter describes the technique used by this company to determine retained austenite.

"MRC uses the integrated intensities method of determining retained austenite on a GE XRD 5 Diffractometer using a CA 7H X-ray tube and a #6 SPG counter tube.

Table 5

Chemical Analysis of Materials Used in

Bearing Components for NASA 3-7.261

	C	Mn	P	S	Si	Cr	Mo	V	Ni	W	Co	Fe	Cu
CVM-M-50 Ring Material (1)	.802	.24	.008	.003	.18	3.95	4.36	.93	-	-	-	Bal.	
GVM-M-50 Ball Material (2)	.802	.24	.008	.003	.18	3.95	4.36	.93	-	-	-	Bal.	
CVM-WB-49 (3)	1.06	.44	.004	.008	.32	4.22	3.76	1.86	.08	6.82	5.29	Bal.	
M-1 (4)	.798	.28	.020	.005	.29	3.74	8.71	1.11	-	1.53	-	Bal.	
S-Monel (5)	.095	.74	-	-	3.69	-	-	-	65.28	-	-	.66	28.78

(1) Allegheny Ludlum Steel Co. Heat #25431

(2) CVM Billet #A - CVM Billet #B

(3) Crucible Steel Co. Heat #07833

(4) Allegheny Ludlum Steel Co. Heat #W81154

(5) Janney Cylinder Co. Heat #AA1(6755)

Table 6

*

Hardness Measurements

Outer Rings (CVM-M-50)

Bearing Number				Bearing Number				Bearing Number			
1.	63.0	63.0	63.0	37.	63.0	63.0	63.1	73.	63.0	63.2	63.0
2.	63.1	63.0	63.0	38.	63.0	63.0	63.0	74.	63.0	63.0	63.0
3.	63.0	63.0	63.2	39.	63.3	63.0	63.0	75.	63.0	63.0	63.0
4.	63.0	63.0	63.0	40.	63.0	63.0	63.1	76.	63.0	63.0	63.0
5.	63.1	63.1	63.0	41.	62.9	63.0	63.0	77.	63.0	63.0	63.0
6.	63.0	63.0	63.0	42.	63.0	63.2	63.0	78.	63.0	63.0	63.0
7.	63.2	63.0	63.1	43.	63.0	63.4	63.0	79.	63.0	63.0	63.0
8.	63.0	63.0	63.1	44.	62.9	62.9	63.0	80.	63.0	63.2	63.1
9.	63.0	63.0	63.0	45.	63.0	63.0	63.0	81.	63.0	63.0	63.1
10.	63.0	63.1	63.1	46,	63.0	63.0	62.9	82.	63.0	63.0	63.2
11,	62.9	63.3	63.1	47.	63.0	63.0	63.0	83.	63.0	63.0	63.0
12.	63.1	63.0	63.1	48.	63.0	63.0	63.0	84.	63.2	63.0	63.3
13.	63.0	63.1	63.1	49.	63.0	63.0	63.0	85.	63.0	63.0	63.0
14.	63.0	63.1	63.0	50.	63.0	63.0	63.0	86.	63.0	63.0	63.0
15.	63.0	63.0	63.1	51.	63.0	63.0	63.0	87.	63.0	63.0	63.0
16.	63.0	63.0	63.0	52,	63.2	63.3	63.4	88.	63.0	63.0	63.0
17.	63.2	63.2	63.1	53.	63.0	63.2	63.0	89.	62.9	63.0	62.9
18.	63.0	62.7	63.0	54.	63.0	63.0	63.0	90.	63.2	63.2	63.0
19.	63.0	63.0	63.0	55.	63.0	63.0	63.1	91.	63.0	62.9	63.0
20.	63.0	63.4	63.1	56.	63.0	63.2	63.2	92.	63.0	63.1	63.1
21.	63.0	63.0	63.0	57.	63.0	63.1	63.0	93.	63.0	63.0	63.0
22.	63.0	63.0	63.0	58,	63.0	63.0	63.2	94.	62.9	63.0	63.0
23.	63.4	63.0	63.2	59.	63.0	63.0	63.0	95.	63.0	63.0	63.0
24.	63.2	63.2	63.1	60.	63.0	63.0	63.1	96.	63.0	63.0	63.0
25.	63.0	63.0	62.9	61.	63.0	63.0	63.0	97.	63.0	63.0	63.0
26.	63.4	63.2	63.2	62.	63.3	63.2	63.2	98.	62.9	63.0	63.0
27.	63.0	63.0	63.1	63.	63.0	63.0	63.0	99.	63.0	63.0	63.0
28.	63.2	63.3	63.1	64.	63.0	63.0	63.0	100.	63.0	63.0	63.0
29.	63.0	63.0	63.1	65.	63.0	63.2	63.1	101.	63.0	63.0	63.2
30.	63.0	63.0	62.9	66.	63.0	63.0	63.0	102.	63.0	63.0	63.6
31.	63.0	63.0	63.0	67.	63.0	63.0	63.0	103,	63.0	63.1	63.1
32.	63.0	63.1	63.0	68.	63.2	63.1	63.1	104.	63.0	63.0	63.0
33.	63.0	63.0	63.0	69.	63.0	63.0	63.0	105.	63.0	63.0	63.0
34.	63.0	63.0	63.0	70.	62.9	63.1	63.0	106.	63.0	63.0	63.0
35.	63.0	63.0	63.0	71.	63.0	63.0	63.0	107.	63.0	63.0	63,0
36.	63.0	63.0	63.0	72.	63.0	63.0	63.0	108.	63.0	62.9	62.9

* All hardness measurements in Rockwell C

Table 7

Hardness* Measurements

Balls . — (CVM-M-50)

Bearing Number				Bearing Number			
1.	63.0	63.0	63.0	37.	63.0	63.2	63.0
2.	63.0	63.0	63.0	38.	63.5	63.5	63.1
3.	63.0	63.0	63.2	39,	63.0	63.4	63.5
4.	63.0	63.0	63.2	40.	63.0	63.0	63.5
5.	63.2	63.2	63.5	41.	63.0	63.0	63.0
6.	63.0	63.0	63.1	42.	63.0	63.0	63.0
7.	63.5	63.4	63.0	43.	63.0	63.0	63.0
8.	63.0	63.5	63.2	44.	63.0	63.1	63.0
9.	63.0	63.0	63.4	45.	63.0	63.0	63.3
10.	63.0	63.0	63.0	46.	63.0	63.0	63.2
11.	63.0	63.2	63.1	47.	63.0	63.0	63.0
12.	63.0	63.0	63.0	48.	63.0	63.0	63.5
13.	63.0	63.0	63.0	49.	63.5	63.0	63.0
14.	63.0	63.0	63.0	50.	63.0	63.4	63.0
15.	63.0	63.5	63.2	51.	63.0	63.1	63.2
16.	63.2	63.2	63.0	52.	63.4	63.0	63.0
17.	63.0	63.0	63.0	53.	63.0	63.0	62.9
18.	63.0	63.0	63.0	54.	63.0	63.0	63.0
19.	63.0	63.0	63.5	55.	63.0	63.0	63.1
20,	63.0	63.0	63.0	56.	63.0	63.3	63.0
21.	63.0	63.0	63.5	57.	63.0	63.1	63.5
22.	63.0	63.0	63.0	58.	63.5	63.5	63.2
23.	63.0	63.Q	63.1	59.	63.0	63.0	63.0
24.	63.0	63.0	63.0	60.	63.2	63.3	63.3
25.	63.0	63.3	63.1	61.	63.2	63.0	63.4
26.	63.1	63.0	63.0	62.	63.1	63.0	63.0
27.	63.0	63.0	63.0	63.	63.0	63.5	63.1
28.	62.9	63.0	63.3	64.	63.0	63.0	63.0
29.	63.0	63.0	63.2	65.	63.0	63.0	63.0
30.	63.0	63.4	63.1	66.	63.5	63.4	63.5
31.	63.2	63.0	63.4	67.	63.0	63.0	63.0
32.	63.0	63.0	63.0	68.	63.0	63.4	63.1
33.	63.1	63.0	63.0	69.	63.2	63.0	63.0
34.	63.0	63.2	63.0	70.	63.5	63.1	63.0
35.	63.0	63.0	63.0	71,	63.2	63.0	63.0
36.	63.0	63.0	63.0	72.	63.3	63.0	63.0

* All hardness measurements in Rockwell C

Table 8

*

Hardness Measurements

Puller Grooved Inner Rings

(CVM M-50)

Bearing Number				Bearing Number				Bearing Number			
1.	63.0	63.5	63.1	38.	62.9	63.0	62.9	75.	63.0	63.0	63.0
2.	63.0	63.0	63.2	39.	63.2	63.0	63.0	76.	63.0	62.9	63.0
3.	63.0	63.0	63.0	40.	63.0	63.0	63.2	77.	63.0	63.0	63.0
4.	63.0	63.0	63.0	41.	63.2	63.3	63.0	78.	63.0	63.2	63.0
5.	63.2	63.5	63.0	42.	63.0	63.0	63.0	79.	63.2	63.4	63.2
6.	63.0	63.0	63.4	43.	63.0	63.0	63.0	80.	63.0	63.0	63.1
7.	63.2	63.2	63.0	44.	63.1	63.0	63.4	81.	63.0	63.0	63.0
8.	63.0	63.2	63.0	45.	63.0	63.0	63.3	82.	63.0	63.0	62.9
9.	65.5	63.0	63.0	46.	63.0	63.0	63.0	83.	63.0	63.0	63.3
10.	63.0	63.2	63.0	47.	62.9	62.9	63.1	84.	63.0	63.0	63.0
11.	63.0	63.0	63.3	48.	63.2	63.0	63.0	85.	63.0	63.0	63.0
12.	63.0	63.0	63.0	49.	63.0	62.9	63.3	86.	63.0	63.0	63.0
13.	63.0	62.9	63.2	50.	63.0	63.1	63.0	87.	63.0	63.0	63.5
14.	63.0	63.1	63.4	51.	63.0	63.0	63.0	88.	63.0	63.0	63.0
15.	63.0	62.9	63.1	52.	63.0	63.0	62.9	89.	63.1	63.0	63.0
16.	63.0	63.0	63.0	53.	63.2	63.5	63.3	90.	63.0	63.0	63.0
17.	63.2	63.0	63.0	54.	63.0	63.0	63.0	91.	63.0	63.0	63.5
18,	63.0	63.2	63.1	55.	63.0	63.0	63.0	92.	63.0	63.0	63.4
19.	63.0	63.0	62.9	56.	63.0	63.1	63.0	93.	63.0	63.0	63.1
20.	63.0	63.5	63.2	57.	63.2	63.0	63.0	94.	63.0	63.0	63.3
21.	63.0	63.0	63.0	58.	63.0	63.0	63.2	95.	63.0	63.2	63.2
22.	63.0	63.0	63.0	59.	63.0	63.0	63.5	96.	63.0	63.0	630
23.	63.0	62.9	62.9	60.	63.0	63.5	63.2	97.	63.0	63.0	63.0
24.	63.0	63.0	63.0	61.	63.0	63.0	63.0	98.	62.9	63.0	63.0
25.	63.0	63.0	63.0	62.	63.0	63.0	62.9	99.	63.0	63.0	63.3
26.	63.0	62.9	62.9	63.	63.2	63.2	63.0	100.	63.0	63.0	63.0
27.	63.1	63.3	63.0	64.	63.2	63.0	63.0	101.	63.0	63.0	63.2
28.	63.0	63.0	63.0	65.	63.5	63.1	63.2	102.	63.0	63.0	62.9
29.	63.0	63.0	63.0	66.	63.0	63.0	63.2	103.	63.4	63.0	62.9
30.	63.0	63.0	63.0	67.	63.2	63.4	63.1	104.	63.4	63.0	63.3
31.	63.2	63.2	63.0	68.	63.0	63.0	63.0	105.	63.2	63.2	63.0
32.	63.0	63.0	63.0	69.	63.0	63.0	62.9	106.	62.9	63.0	62.9
33.	63.0	63.2	63.1	70.	63.0	63.0	63.0	107.	63.1	63.0	63.2
34.	63.0	63.2	63.1	71.	63.0	63.0	63.0	108.	63.3	63.0	63.0
35.	63.1	63.0	63.4	72.	62.9	62.9	63.0	109.	63.2	63.0	63.3
36.	63.0	63.0	63.0	73.	63.0	63.0	63.1	110.	63.0	63.0	63.0
37.	63.0	63.2	63.1	74.	63.0	63.1	63.2	111.	63.0	63.2	63.0

* All hardness measurements in Rockwell C

Table 9

Hardness Measurements

Inner Rings (WB–49)

Bearing Number				Bearing Number				Bearing Number			
1.	64.9	65.0	64.9	33.	64.9	64.9	64.5	65.	65.0	65.0	65.0
2.	65.0	64.9	65.0	34.	64.8	64.9	64.8	66.	65.0	65.0	65.1
3.	65.0	65.0	65.0	35.	65.0	65.0	64.8	67.	65.0	65.0	65.0
4.	65.0	65.0	64.9	36.	65.0	65.0	65.0	68.	64.9	65.0	65.0
5.	65.0	65.0	65.0	37.	64.9	65.0	65.0	69.	65.0	64.9	65.0
6.	65.0	65.0	65.0	38.	65.0	64.9	65.0	70.	65.0	65.0	64.9
7.	65.0	65.0	64.9	39.	65.0	65.0	65.0	71.	64.9	64.8	64.8
8.	65.0	65.0	65.0	40.	65.0	64.8	65.0	72.	65.0	65.0	64.9
9.	65.0	65.0	65.0	41.	65.0	65.0	65.0	73.	65.0	64.8	65.0
10.	65.0	64.8	65.0	42.	64.9	64.9	64.6	74.	65.0	65.0	65.0
11.	65.0	65.0	65.0	43.	65.0	64.9	65.0	75,	65.0	64.9	65.0
12.	64.9	64.8	65.0	44.	65.0	65.0	65.0	76.	64.8	65.0	65.0
13.	65.0	64.9	65.0	45.	64.9	64.8	65.0	77.	65.0	65.0	64.9
14.	65.0	65.0	65.0	46.	64.9	64.9	64.9	78.	65.0	64.9	65.0
15.	64.8	65.0	64.8	47.	65.0	64.9	65.0	79.	65.0	64.8	65.0
16.	64.9	65.1	65.0	48.	64.9	64.9	65.0	80.	65.0	65.0	64.9
17.	65.0	64.9	64.7	49.	64.8	65.0	65.0	81.	65.0	65.0	64.8
18.	65.0	64.8	65.0	50.	65.0	65.0	65.0	82.	65.0	65.0	65.0
19.	65.0	65.0	65.0	51.	64.8	65.0	65.0	83,	64.8	65.0	65.0
20.	65.0	65.0	65.0	52.	64.9	65.0	65.0	84.	65.0	65.0	64.7
21.	65.0	65.0	64.8	53.	65.0	65.0	65.0	85.	65.0	65.0	64.9
22.	64.9	64.9	65.0	54.	64.9	65.0	65.0	86.	65.1	64.6	65.0
23.	65.0	64.9	65.0	55.	65.0	65.0	64.9	87.	65.0	64.8	65.0
24.	65.0	65.0	65.0	56.	64.9	64.9	64.8	88.	65.0	65.1	65.2
25.	65.0	65.1	65.0	57.	65.0	65.0	65.0	89.	64.7	64.9	65.0
26.	65.0	65.0	64.9	58.	64.9	65.0	64.9	90.	64.7	64.9	65.0
27.	65.0	65.0	64.9	59.	64.9	64.9	64.6	91.	64.5	64.8	64.9
28.	65.0	65.0	65.0	60.	64.9	64.9	64.9	92.	65.0	65.0	65.1
29.	65.0	65.0	65.0	61.	65.0	64.9	65.0	93.	64.9	65.0	65.2
30.	65.0	65.0	64.8	62.	64.8	64.8	65.0	94.	64.5	64.9	64.9
31.	65.0	65.0	64.9	63.	65.0	65.0	65.0	95.	65.0	65.0	65.2
32.	64.9	65.0	64.9	64.	65.0	65.0	64.8				

* All hardnesses in Rockwell C

Table 10

*
Hardness Measurements

Outer Rings (WB–49)

Bearing Number				Bearing Number				Bearing Number			
1.	65.4	65.2	65,0	31.	65.0	65.0	65.0	61.	65.0	65.0	65.0
2.	65.0	65.0	65.0	32.	65.2	65.0	65.0	62.	65.0	65.0	65.0
3.	65.0	65.0	65.2	33.	65.0	65.1	65.0	63.	65.2	65.0	65.1
4.	65.0	65.0	65.0	34.	65.0	65.1	65.0	64.	65.1	65.1	65.0
5.	65.0	65.0	65.0	35.	65.0	65.0	65.0	65.	65.0	65.0	65.0
6.	65.0	65.0	65.0	36.	65.2	65.0	65.1	66.	65.1	65.0	65.0
7.	65.0	65.0	65.0	37.	65.1	65.2	65.2	67.	65.0	65.1	65.1
8.	65.0	65,0	65.0	38.	65.0	65.0	64.9	68.	65.1	65.1	65.0
9.	65.0	65.0	65.0	39.	65.0	65.0	65.0	69.	65.0	65.0	65.0
10,	64.8	65.0	65.0	40.	65.0	65.0	65.0	70.	64.8	64.6	64.8
11.	65.0	65.0	65.0	41.	65.1	65.0	65.3	71.	64.8	64.8	64.8
12.	65.0	65.0	65.1	42.	65.5	65.3	65.0	72.	65.0	64.8	64.9
13.	66.0	65.0	65.0	43.	65.0	65.1	65.2	73.	65.0	64.7	64.8
14.	65.0	65.0	65.0	44.	65.0	65.0	65.0	74.	64.8	64.9	64.9
15.	65.0	65.0	65.0	45.	64.5	64.5	65.0	75.	64.9	64.8	65.0
16.	65.2	65.2	65.0	46.	65.0	65.2	65.2	76.	64.8	64.8	64.8
17.	65.1	65.0	65.0	47.	65.1	65.3	65.0	77.	64.7	64.3	65.0
18,	65.0	65.0	64.8	48.	65.0	65.2	65.0	78.	64.8	64.5	65.0
19.	65.1	65.0	64.9	49.	65.0	65.0	65.0	79.	64.9	64.9	64.9
20.	65.0	65.0	65.1	50.	65.0	65.2	65.1	80.	64.7	64.7	65.0
21.	65.2	65.2	65.0	51.	65.1	65.0	65.0	81.	64.5	64.9	64.9
22.	65.0	65.1	65.1	52.	65.0	65.0	65.0	82.	64.7	64.8	64.7
23.	65.0	65.0	65.1	53.	65.0	65.1	65.0	83.	64.9	64.8	64.8
24.	65.1	65.0	65.0	54.	65.1	65.0	65.2	84.	64.8	64.8	64.8
25.	65.2	66.4	65.1	55.	65.2	65.2	65.1	85.	64.8	64.9	65.0
26.	65,0	6 ,0	65.0	56.	65.2	65.0	65.0	86.	65.0	64.6	64.7
27.	65.2	65.0	65.1	57.	65.0	65.1	65.0	87.	64.8	64.9	64.9
28.	65.0	65.0	65.0	58.	65.0	65.0	65.1	88.	64.8	64.9	64.7
29.	65.0	65.0	65.0	59.	65.0	65.0	65.0	89.	64.8	64.8	64.9
30.	65.0	65.0	64.9	60.	65.1	65.0	65.0				

* All hardnesses in Rockwell C

Surface preparation is particularly important to assure that no heat treating or grinding conditions affect the measurements. MRC electro-polishes all surfaces to be tested. A minimum depth of .001" is used for ground parts, .020" for unground surfaces. Other methods of preparation are not acceptable as false readings will always be found.

Parts are tested at the following machine settings:

Chrome Radiation 50 KVP 30 MA
3° Medium Resolution Beam Collimator
Medium Resolution Detector Collimator
.0015" Vanadium Filter
500 Count Per Second on Rate Meter
2 Second Time Constant
Goniometer Speed 2° per Minute
Chart Speed 60" per Hour

M-50 steel is scanned for the 200 austenite line from 20 74° through 20 83°, the peak occurring at about 79.4' 20, The martensite 200 peak is scanned between 20 98' and 20 114°, the peak being found at about 106° 20. A further scan may be made for the 220 austenite line between 20 122' and 20 134°.

The chart is placed on a desk and a straight line drawn through the background radiation. The area above the line and under the peaks is measured with a planimeter and put in the formula:

$$\text{Ret, Aust.} = \frac{1.0 - .04\ (\text{Carbide Correction})}{\left(\frac{\text{Area Martensite 200}}{\text{Area Austenite 200}} \times 1.74\right) + 1}$$

$$\text{or } V = \frac{1}{1 + \frac{200a}{200r}}$$
$$.576$$

In general production, a scan for the austenite 200 line is sufficient to determine if a peak is present. No peak at this location indicates a retained austenite content of less than 5%."

Using the above technique, the retained austenite content was determined to be less than 3% in the ring and ball material.

4.0 Test Facilities

To perform the testing phase of this program, two major test facilities were required. Of these, the rolling contact fatigue testers (RC Rig) were available and could be immediately utilized. The full scale bearing testers, however, had to be designed, constructed and installed and checked out. Both of these facilities will be described in the following paragraphs.

4.1 Rolling Contact Fatigue Testing (RC Rig)

To determine the relative effects of the three experimental lubricants on the selected bearing materials, Rolling Contact Fatigue tests were performed prior to initiation of the full scale bearing tests. Testing was also conducted to determine the differences to be expected between the two major bearing materials (M-50, WB-49) and between the two CVM heats of M-50 used in the subject program.

The RC Rig, briefly described below, became a major tool in the pre-bearing test evaluation of the three fluids and two bearing materials. A view of the RC Rig is shown in Figure 3 and a close-up of the test bar roller geometry is shown in Figure 4. The cylindrical bar, three inches long by .375" D., which is the test specimen, drives two hemi-spherically ground rollers. These rollers are each 7 1/2" D. and have a contact radius of 1/4". They are mounted via double row ball bearings and are supported by a massive pendulum yoke. Load is applied by closing the rollers together against the test bar. The loading train consists of a micrometer threaded turnbuckle and a calibrated load cell, The load cell is a four strain gage bridge which feeds into a Baldwin SR4 strain analyzer. The load cells are calibrated using a tensile machine and obtaining a load versus micro-inch deflection curve. The rollers impinging upon the test bar produce a compressive stress pattern (Hertzian stress) very similar to that seen on the inner ring of a ball bearing. The resultant spalling fatigue failure is essentially identical to that observed on actual bearing races as shown in Figure 5. Lubrication is by a drip feed system, a needle valve being used to control the flow rate

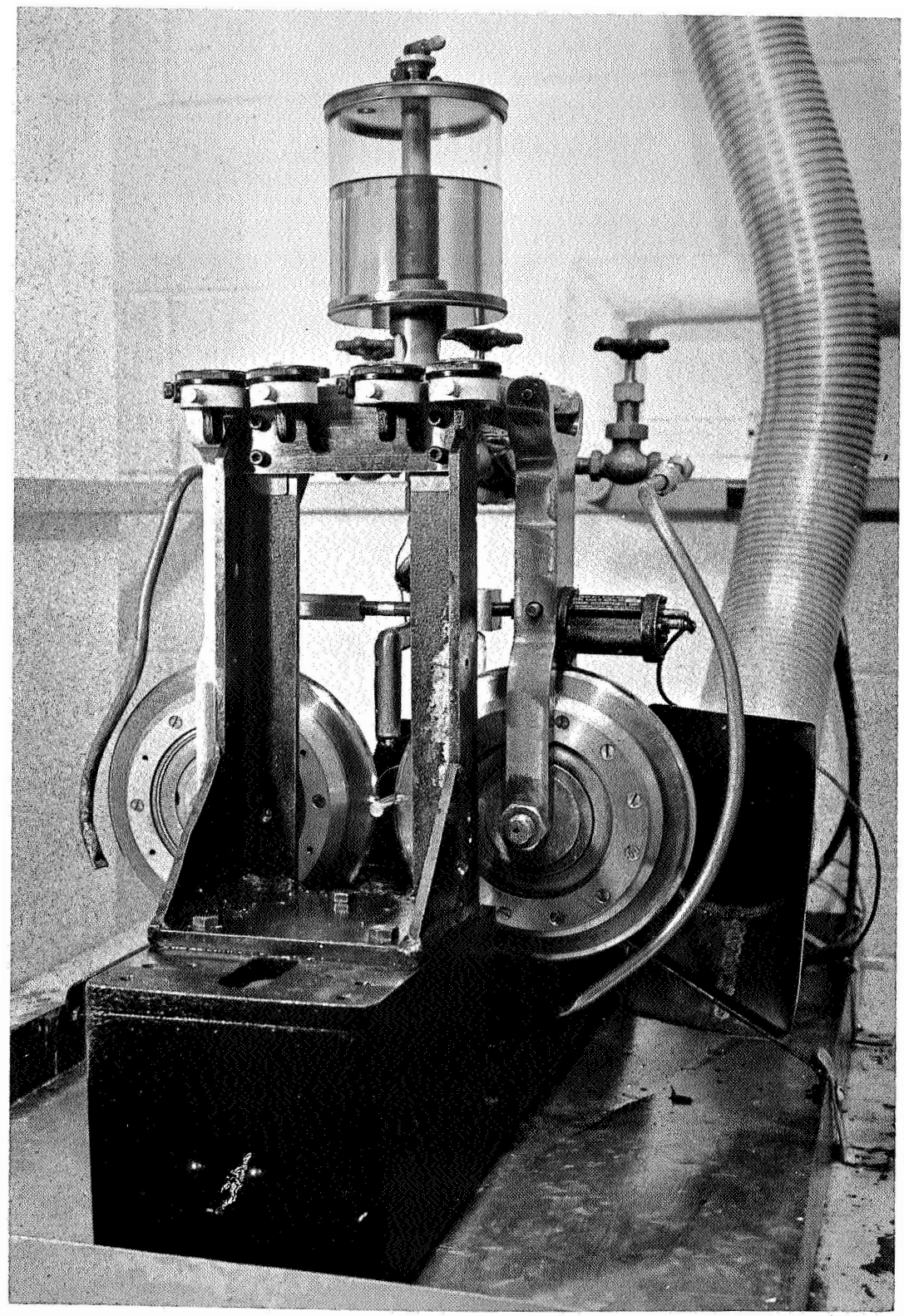
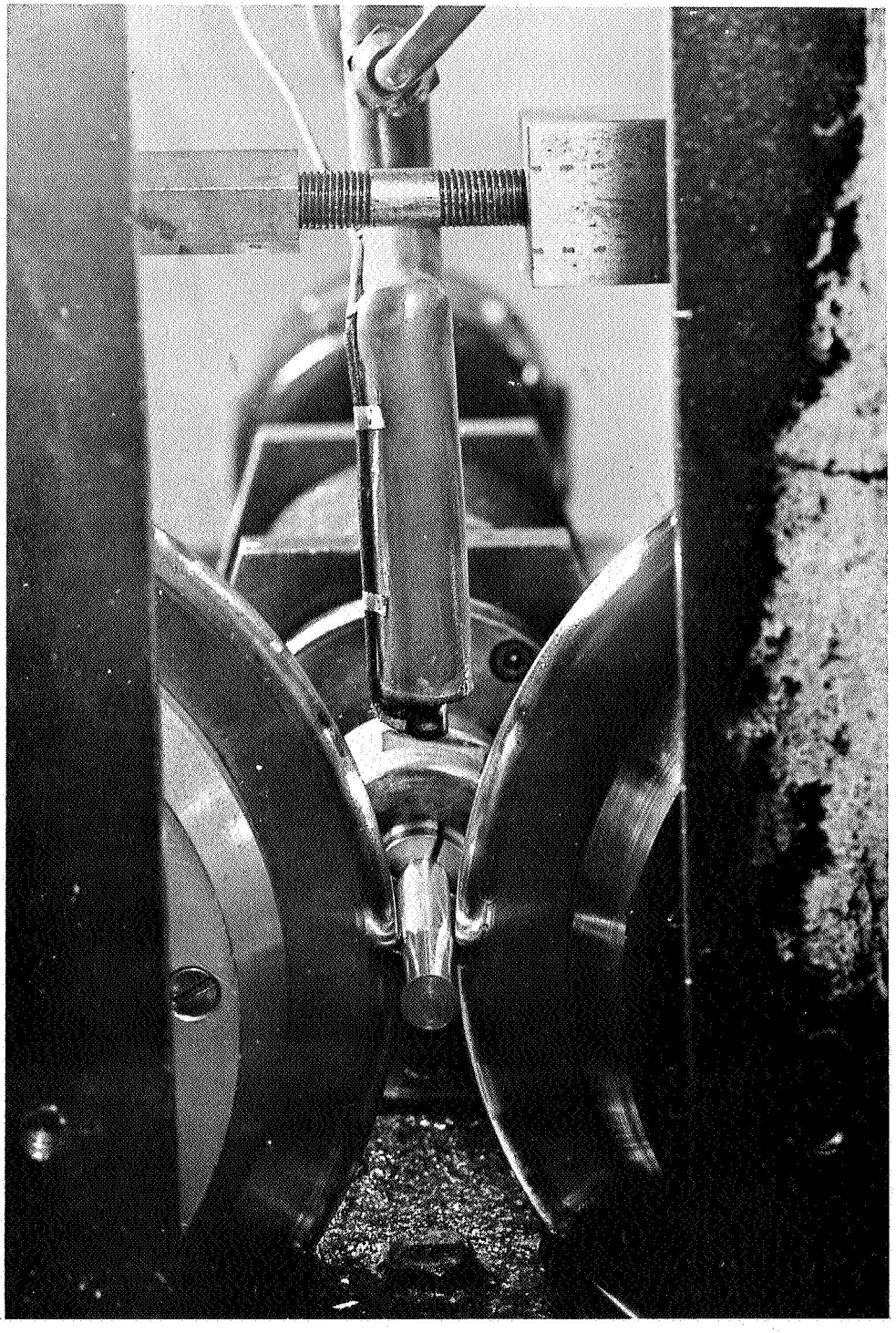


FIGURE 3. GENERAL ELECTRIC RC RIG (C65062510)



FIGURF: 4. CLOSE-UP OF RC TEST BAR - ROLLER GEOMETRY (C65062512)

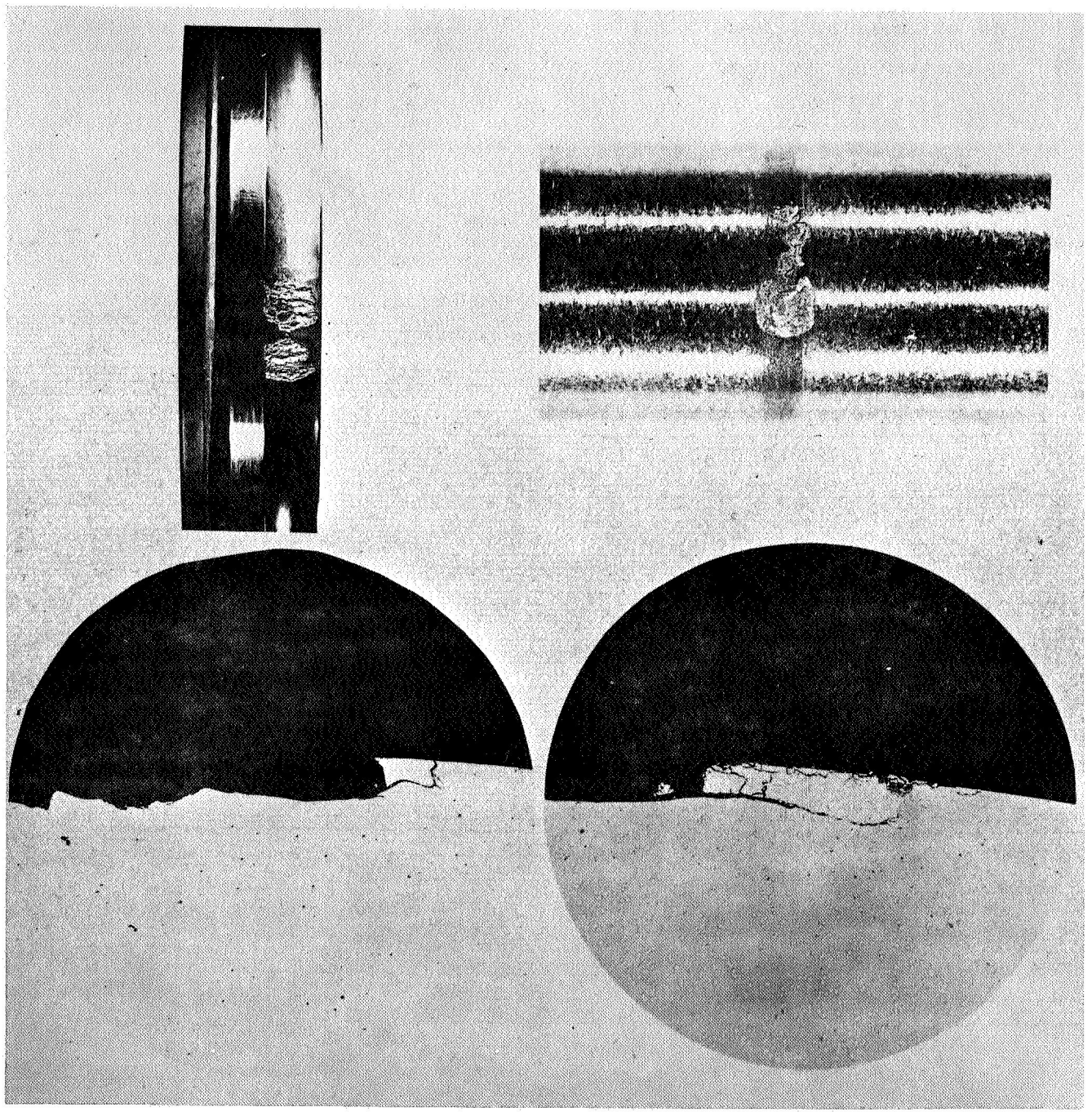


FIGURE 5. FAILURE SIMILARITY BETWEEN ACTUAL BEARING AND RC RIG SPECIMEN

TOP LEFT:	FATIGUE SPALL ON INNER RING OF BEARING.
TOP RIGHT:	FATIGUE SPALL ON RC RIG BAR.
BOTTOM LEFT:	PHOTOMICROGRAPH OF FATIGUE SPALL ON BEARING.
BOTTOM RIGHT:	PHOTOMICROGRAPH OF FATIGUE SPALL ON RC RIG BAR.

at a constant rate of 20 drops per minute. Studies have shown that this flow rate is sufficient to provide adequate lubrication as evidenced by the oil meniscus, seen in Figure 6. A velocity type vibration pick-up is mounted on one of the support yokes. This instrument acts as a failure sensor and terminates the test immediately upon occurrence of a spalling fatigue failure.

In the current program, the test bars were heated via a high frequency induction coil wrapped around the test bar both fore and aft of the contacting rollers. An "Ircon" Infra-red Pyrometer trained on the contact area measures and controls the temperature. Exact temperature control is achieved by using a pyrometer controlled saturable reactor control to cycle the induction heater, Figure 7 shows a view of the Pyrometer Control set-up and a close-up of the induction coil is shown in Figure 8,

The oil is heated by passing it through a hot air heat exchanger prior to dropping it onto the bar. A thermocouple at the outlet of the heat exchanger section measures the oil drop prior to its release, This is shown clearly in Figure 4. Lastly, the rollers are heated by a hot air manifold around a portion of the rollers. This is shown in Figure 3, Heating the rollers prevents thermal gradients and helps to achieve and maintain, close temperature control.

4.2 120 mm Bearing Tester

The design of the bearing testers was dictated by the severe environmental conditions expected during the test, Among these were:

- Continued and prolonged high speed operation at 600F,
- Maintenance of N2 atmosphere having less than 5 ppm of O2'
- Exposure to potentially corrosive fluids,

The first of the above listed conditions required that all tester components requiring absolute dimensional integrity over the duration of the test

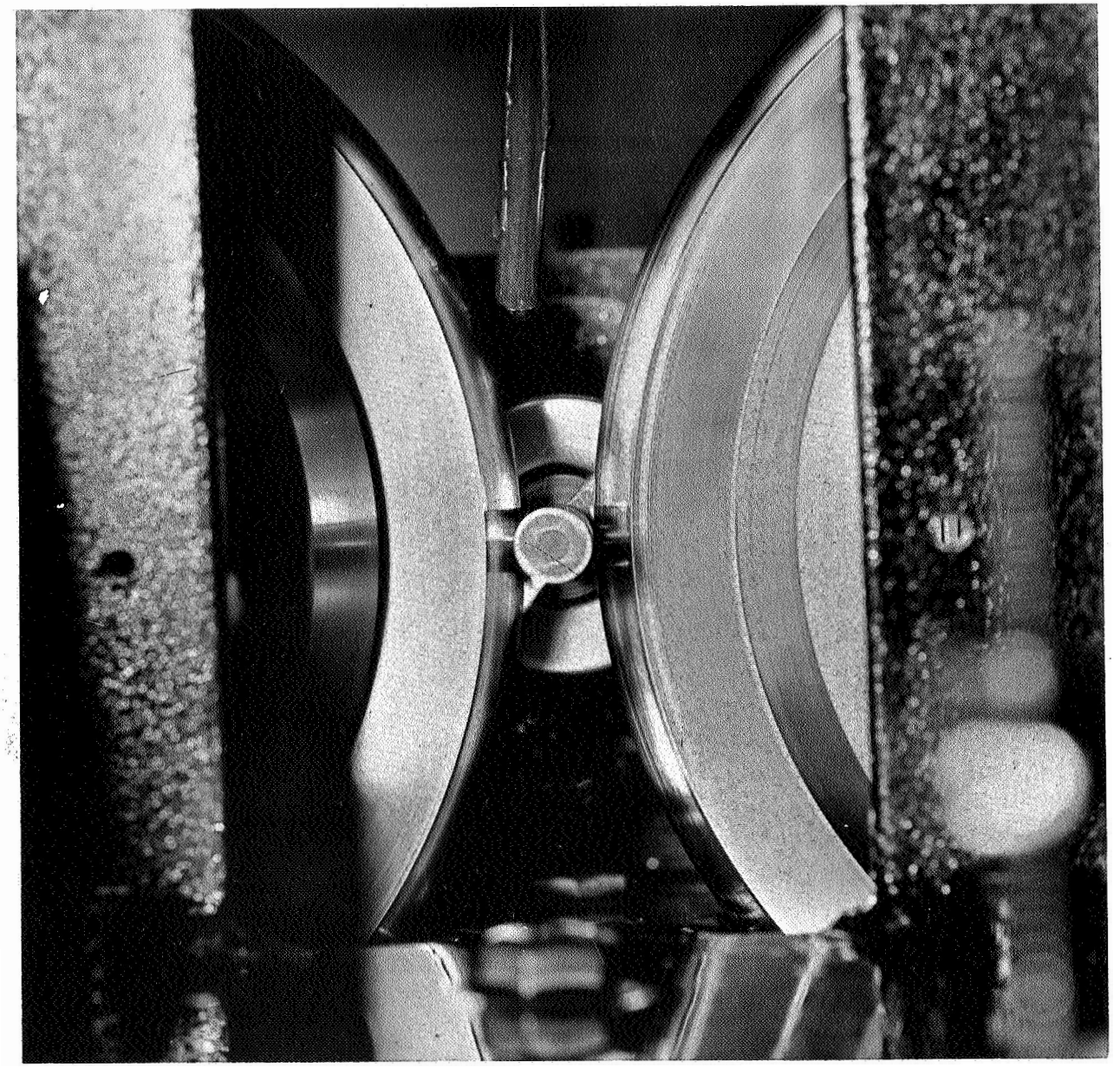


FIGURE 6. HIGH SPEED PHOTOGRAPH OF RC RIG TEST BAR AND ROLLERS. NOTE OIL MENISCUS BETWEEN BAR AND ROLLERS INDICATING PROPER LEVEL OF LUBRICATION.

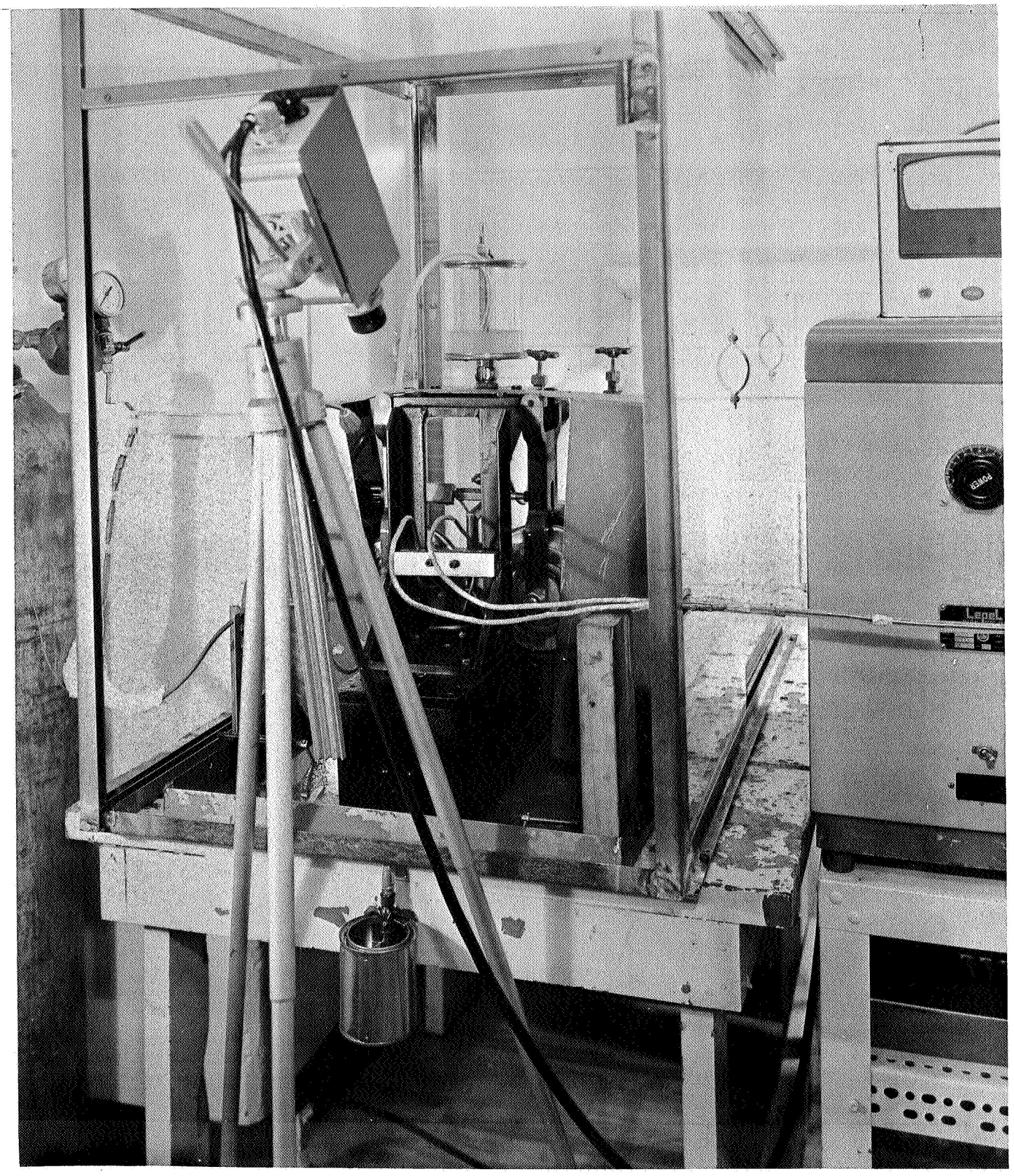


FIGURE 7. INFRARED PYROMETER IN USE ON RC RIG

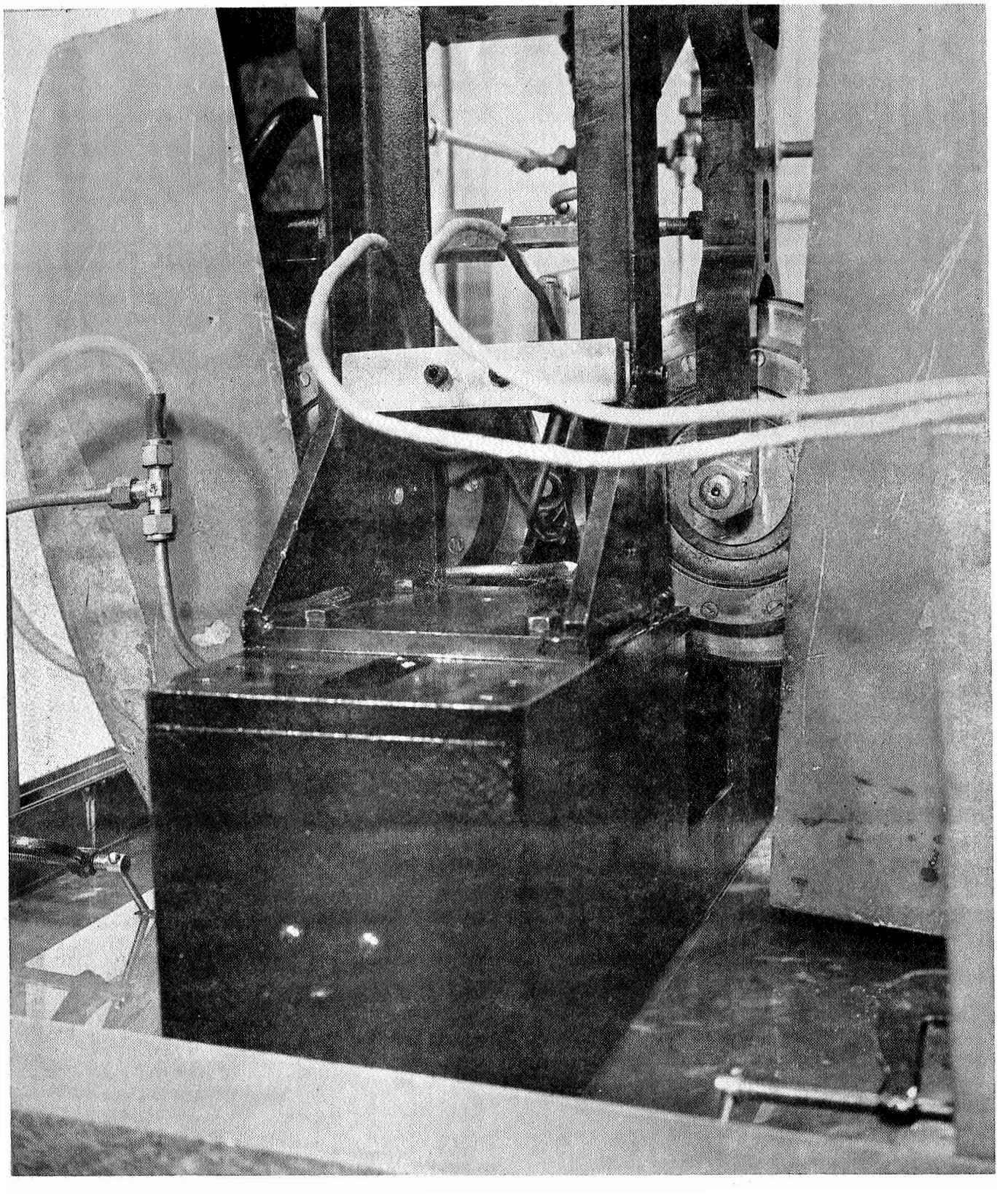


FIGURE 8. HIGH FREQUENCY INDUCTION COILS ON RC RIG.

period had to be made out of a material having good creep properties in the temperature range of interest. AISI 4340 and 4140 were selected and heat treated to assure dimensional stability to temperatures well above the expected maximum operating temperature of 600F.

The N_2 atmosphere was maintained by creating a 5 psi positive pressure in the test cavity. This was achieved by using the tandem carbon seal (Figure 9) used in the #1 seal position on the General Electric J79 turbo jet engine. The N_2 gas is introduced between the two seal elements and thus simultaneously pressurizes the seal as well as inerts the test cavity. Since the test cavity and the test oil sump are an integral sealed system, inertion is simultaneously maintained on the test oil reservoir. To maintain a positive flow a vent line from the oil tank allows overboarding of exhaust fumes. The inerting system described, above was totally successful *as* will be seen later in the section on test results.

The potential corrosiveness of the fluorinated polymer had been cited[3] and consequently steps were taken to negate any potential effects on the bearing tests caused by the reaction of this material with tester components. A metallurgical investigation was performed by the Materials Development Laboratory (MDLO) in which samples of materials under consideration for the bearing tester structural members were exposed to the PR-143 in air, at 600F for periods up to 44 hours. The results of this examination are shown in Table 11. In addition to noting the vispal appearance of the test samples, metallographic examination of the specimens were also made and showed no evidence of subsurface and/or intergranular attack on either the Ni plated materials or the Monel.

Based on these results, it was decided that all parts in the main tester body subject to contact with the test oil would be given a Nickel plating per AMS Spec. 2404. Plating thickness was .0003" min. Furthermore, all parts which either because of their unsuitability to reliable plating,

		BALTIMORE, MD.			ECN 95234
578C839 P2	SSCY 1888417	STEIN SEAL CO. PHILADELPHIA, PA.	B2	B	1) DELETED CRITICAL SYMBOL FROM NOTE 3 CIDN 111871
			C7 B3	C	1) ADDED 2) ADDED NOTE 7

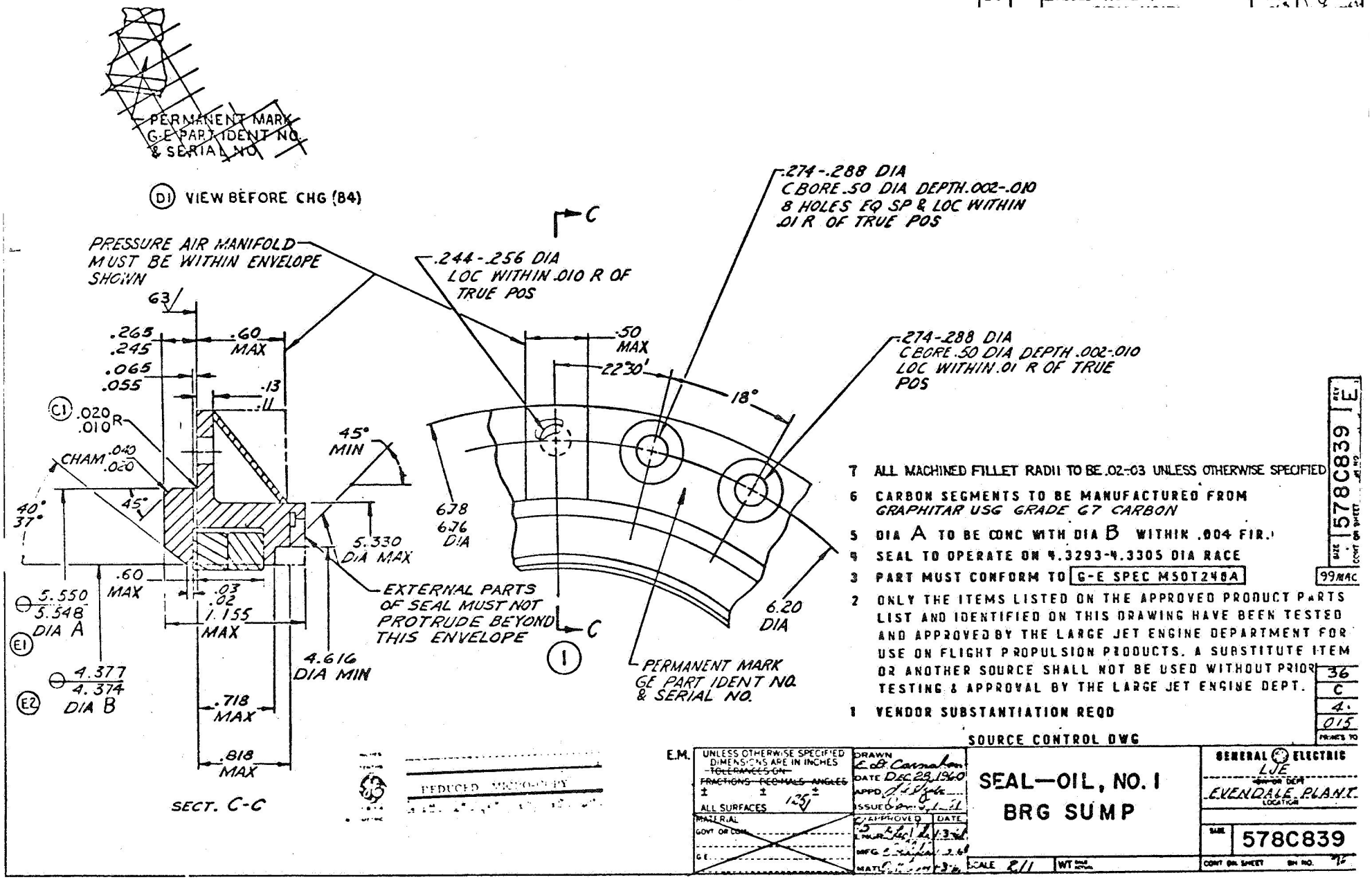


FIGURE 9. NITROGEN SEAL FOR TEST RIG

Table 11

Corrosion Test of Potential

Bearing Test Rig Materials In

DuPont PR-143 (Fluorinated Polymer)

At 600°F Lubricant Temperature

Test Sample Material	Initial Weight	Final Weight	% Changed	Appearance
1. 302 Stainless	7.7273	7,8657 *	+ 1.8	Corroded
2. 302 Stainless	7.8975	8,0324 *	+ 1.7	Some attack
3. Monel	27.2087	27.2718 *	+ 0.2	Very good
4. Monel	27.3449	27.4459 *	+ 0.2	Very slight attac
5. M-50	5.8108	5.8271 *	+ 0.2	Discolored
6. M-50	7.0273	7,0499 *	+ 0.3	Discolored
7. Uncoated Low Carbon Steel (4340)	15.5346	15.4477 **	- 7.	Very corroded flaky rust
8. Nickel Plated(1) Low Carbon Steel (4340)	16.4636	16.4639 **	nil	Excellent
9. Nickel Plated(1) Low Carbon Steel (4140)	$5.9177	15.9177 **	nil	Excellent

The stainless and M-50 samples were sanded with emery cloth to assure complete contact of all surfaces with the PR-143. All samples were then degreased with acetone and dried before weighing initially. The first six samples were started 12:00 noon, 1/4/65, removed at 8:00 a.m., 1/5/65 cleaned with acetone, dried and reweighed (interim weight not shown),

The low carbon steel samples were prepared, degreased with acetone, weighed and all nine samples were put into the 600°F furnace @ 9:15 a.m., 1/5/65.

All samples were removed from furnace @ 9:15, 1/6/65, rinsed with acetone, dried and reweighed (final weight column).

(1) .0003" min thickness, plated per AMS 2404.

* 44 hours immersion

** 24 hours immersion

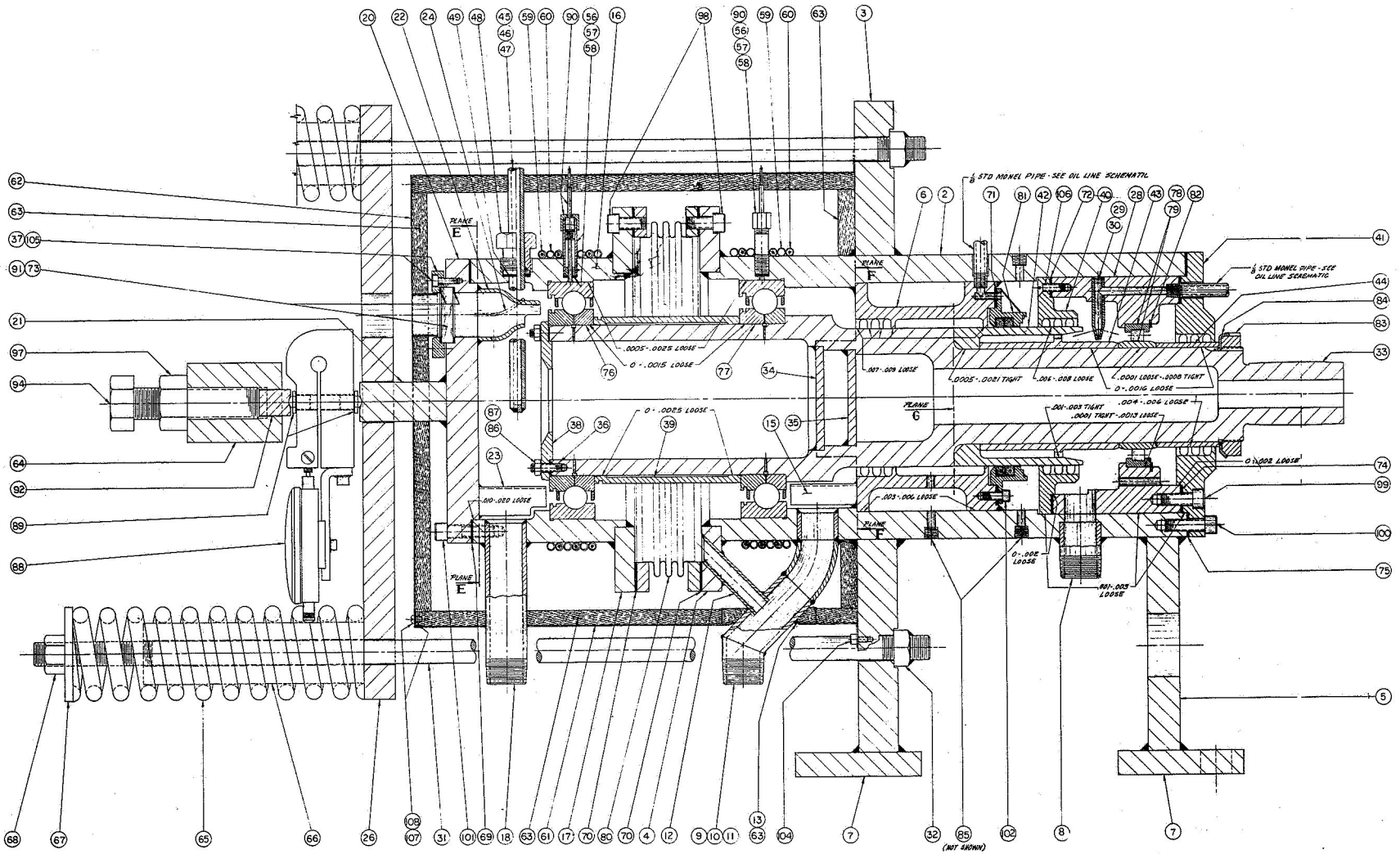
such as the ID of tubes and fittings, or those parts subject to wear, such as pumps, would be made out of Monel. Consequently, the specially designed vane lube pump, the oil tank, all oil supply and return lines were manufactured out of Monel. Also made out of Monel were parts which, because of repeated use would not retain their plating. This included studs, bolt heads and other similar fasteners.

Additionally, compatibility tests were conducted on various elastomers and gasketing materials and it was found that both Viton A and Teflon were acceptable when used with the PR-143.

The test machine assembly is shown in Figure 10. Two stationary housings connected by a bellows form the test sump. One housing is fixed and contains a slave or auxiliary roller bearing to support the test shaft at the drive end; the second housing is free floating and is loaded by a system containing ten springs of 1,000 lb. working capacity each. The test bearings in each housing are axially loaded against each other by the spring system. The load path is from the springs to a circular load plate, through studs, strongback and force-gage to the forward housing. From the forward housing the load is transmitted through the forward test .bearing shaft sleeve and aft test bearing to the aft housing, thence to springs via tie rods which penetrate the load plate. The two test bearings are mounted on a test shaft, and are separated by the sleeve which conducts the axial load between inner races. The test bearings are retained on the shaft by a bolted retaining plate, the shaft is therefore located by the test bearings and has axial freedom through the slave roller bearing. Drive of the test rig is accomplished by a flat belt on a crowed spindle at the roller bearing end of shaft.

Initially, two oil systems were provided. These were:

1. Individual oil jets for each test bearing in conjunction with a single axial jet into the: hollow shaft.



SECTION A-A (COMPOSITE)

FIGURE 10 120 MM BEARING TESTER

2. A single oil jet into the hollow shaft with test bearing oil feed achieved by radial metering holes through the Shaft wall to bearing inner ring split.

In each system approximately three gpm (@ 600F) was assigned for maintaining temperatures and exited from the shaft bore, with an additional three gpm routed through the test bearings. Gravity drainage is provided by a single exit from each sump zone outboard of the test bearings with an auxiliary drain from the sump zone between bearings.

It was found that the oil jet into the hollow shaft was not required for good temperature control and did in fact, contribute to a foaming problem. Consequently, this system was eliminated during the check-out phase of the program.

Sealing between the hot oil sump and the auxiliary bearing sump was achieved by a labyrinth seal and a tandem carbon seal, The latter is nitrogen pressurized to maintain a neutral atmosphere in the test oil system. The tandem carbon seal provides radial sealing against a hollow seal race cooled by oil from the auxiliary bearing oil system.

Lubrication of the auxiliary bearing was provided by a common system for the eight test machines, which feeds two double jets in each tester, 1.0 gpm assigned to the roller bearing, 1.0 gpm for heat removal from carbon seal race.

The hollow shaft contains a double bulkhead between the hot test sump zone and the auxiliary bearing/seal zone. The auxiliary bearing end of the bore is cooled by an air tube projecting through the drive spindle. Auxiliary bearing oil seal is achieved by labyrinth seals.

Test oil circulation was accomplished by a slinger-type pump immersed in the test oil reservoir, discharged through a 100 mesh filter. Figure 11

illustrates the data obtained in evaluating the capacity of this pump at 600F and using both oil delivery methods. Eventually, the oil pressure was set and maintained at 2.7 gpm at 26 psi using only the two oil jets to lubricate the bearing. The test oil reservoir is heated through a salt-bath transfer medium by 6KW strip heaters, thermostatically controlled.

The test shaft is driven at 12,000 rpm by a flat belt from a 40 hp, 3550 rpm motor mounted beneath the test assembly. Safety circuits were provided on each machine, activated by test oil pressure and temperature, test bearing temperature, auxiliary oil pressure and temperature, test bearing fatigue failure (vibration), drive belt failure, and loss of nitrogen pressure. Figure 12 is a schematic of the test stand, drive and hot oil system. Figure 13 is an overall view of the test area.

Instrumentation consisted of:

1. Test bearing temperature achieved by:

   Contact thermocouples at the test bearing outer races, continuously operative.

   Correlative determination of bearing inner-race temperature by optical pyrometer through a sapphire window in the floating housing, for which a shielding tube in the sump and a calibration thermocouple (on the shield) are provided.

2. Vibration

3. Oil Temperature

4. N_2 Dew Point

5. N_2 Flow and Pressure

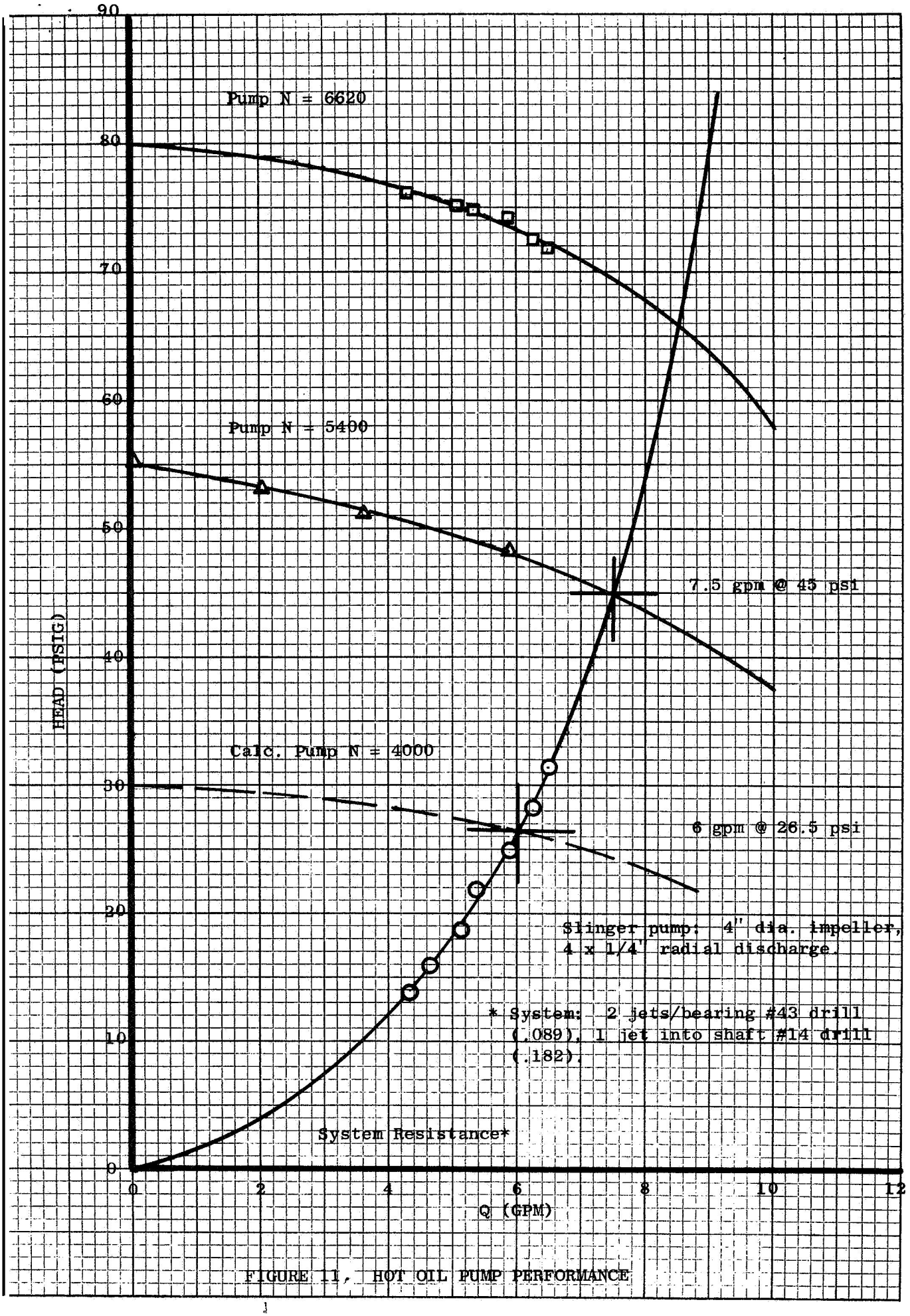


FIGURE 11, HOT OIL PUMP PERFORMANCE

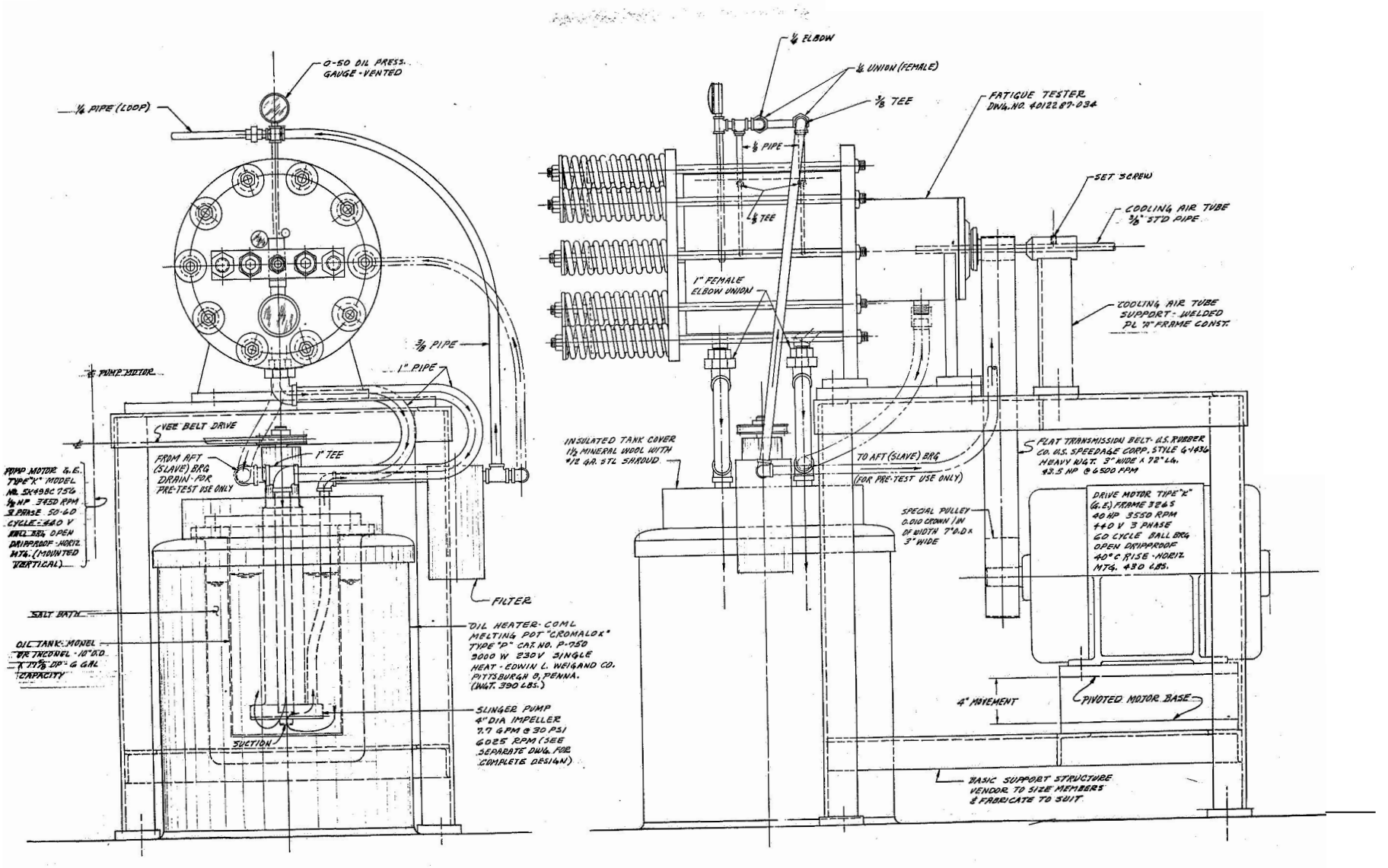


FIGURE 12. DESIGN OF TEST STAND AND LUBRICATION SYSTEM FOR BEARING FATIGUE TESTER

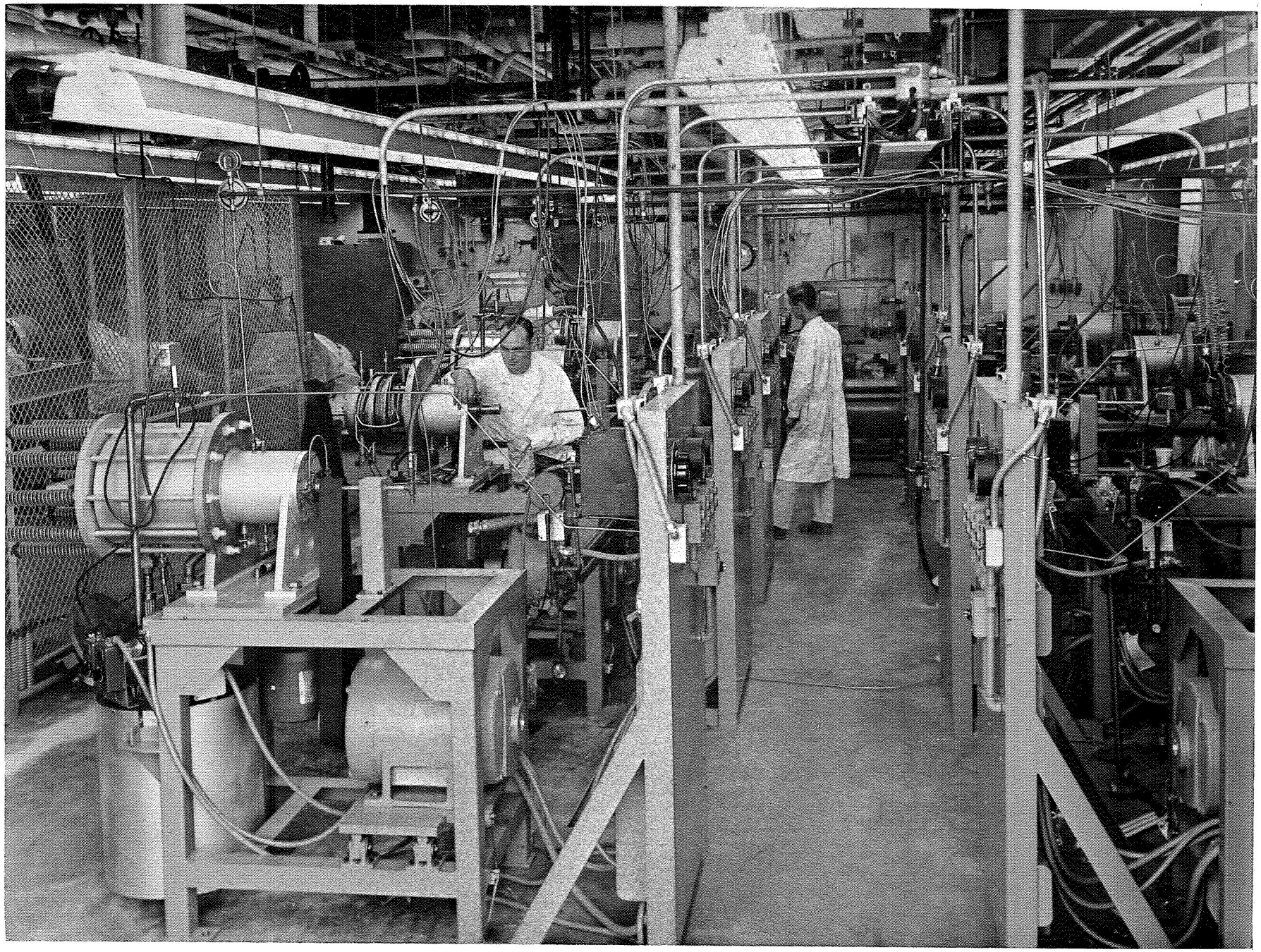


FIGURE 13 VIEW OF BEARING TEST AREA

6. Speed

The use of infra-red pyrometry to measure inner ring temperatures proved to be quite effective. The testers were equipped with two reference points used to calibrate the pyrometer. One of these is a target which is thermocoupled and is just inside the viewing port. The second is the bearing outer ring which is also continually temperature monitored with a thermocouple. By calibrating the pyrometer against these two reference points, and then focusing on the inner ring, a very accurate measurement of inner ring temperature could be made.

Two methods were used to provide continuous monitoring of test bearing performance, A rotary switch of 45-second cycle provides 5-second sensing of vibration pick-up output at each machine plus one 5-second spare station and simultaneously connects the appropriate trip circuits, the latter providing a means of shutting down individual machines by interruption of the common holding circuit through all load contacts. The vibration millivolt output is channeled to give a read-out of displacement and also to energize a limit meter which will trip out the appropriate machine when a manually set limit is exceeded. A fuse to ground is used in each trip circuit to identify shutdowns caused specifically by vibration. A selector switch in the temperature recorder provided trip circuit shut-down of individual machines from test bearing over-temperature, the cycle providing sensing of test bearing temperatures (two bearings per machine) on a 40-second interval.

The determination of proper bearing loading was by means of loading spring deflection and a Dillon force gage. Calibration curves for the springs and the bellows connecting the front and rear bearing housing are shown in Figure 14. Figure 15 is a schematic of the loading system and is included for reference to the assembling and loading procedure described below:

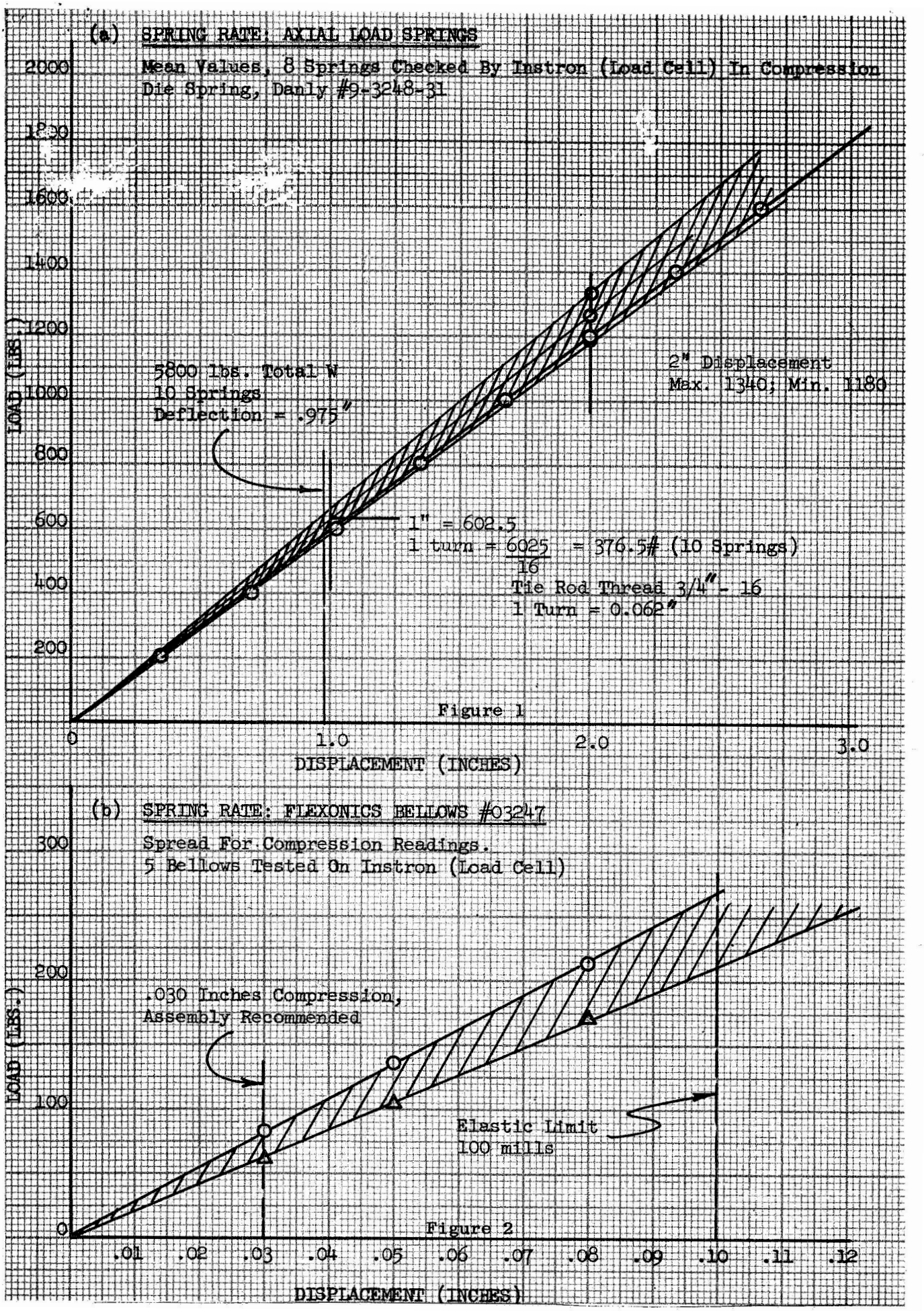


Figure 1

Figure 2

FIGURE 14. CALIBRATION CURVES, BEARING TESTER SPRINGS AND BELLOWS

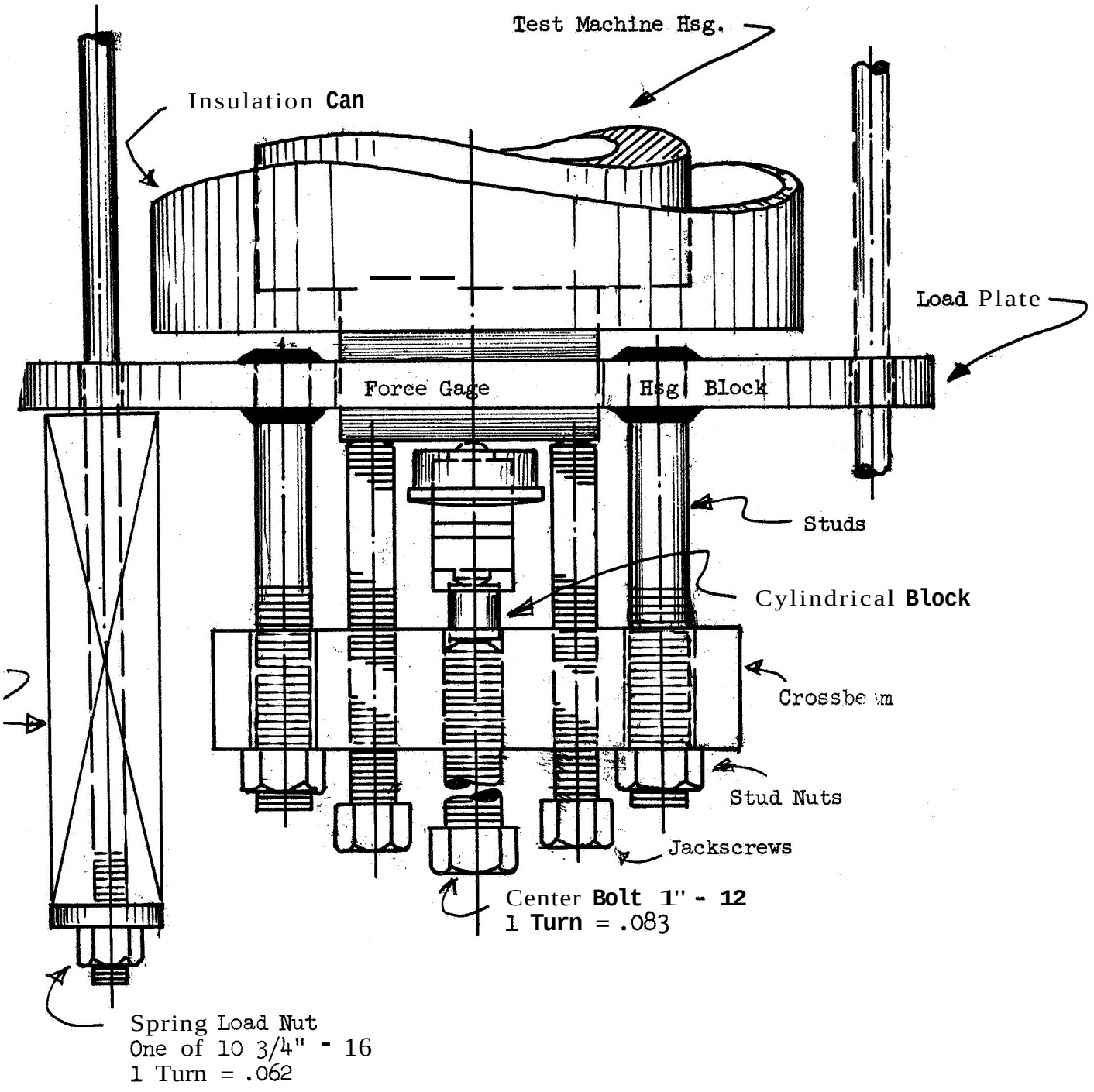


FIGURE 15. SCHEMATIC OF BEARING TESTER LOAD STRUCTURE

1. With floating-housing/bellows flange unbolted, a 10,000 lb. load is applied through the load structure and force gage to align the floating test bearing. The bellows flange bolts are tightened while this load remains applied, so that no radial load component is exerted by the bellows in the final assembly.

2. For 5800 lb. final load setting, the initial 10,000 lb, load is reduced to 725 lb. The load-plate is aligned parallel to the housing face by adjustment of diagonally opposite springs. The cross beam is then aligned parallel to the load-plate, and is shifted within stud clearance to align the force gage load-axis perpendicular to the load-block and load-plate.

3, The gage load is set at 725 lb. by adjustment of the center bolt.

4. The load is transferred to the jackscrews, the gage is removed and a steel ball and spacer block are substituted.

5. Load is transferred to the steel ball,

6. Working load is applied by torqueing the nuts on the ten spring tie-rods in diagonal sequence, a few turns at a time to a total of 14 turns/spring, to give a final load of 725 + (14) (376.5) = 5800 lb.

7. Upon completion of a test run, the load is checked prior to dismantling the test machine. This is done by transferring the full load from the steel ball to the force gage using the jackscrews and center bolt, i.e., by supporting the load on the jackscrews during removal of ball by backing-off center bolt, so that no displacement of the load structure occurs during transfer. Some error due to bolt friction is incurred but results check4 100 lb. and serve to verify test loads.

The inerting nitrogen was supplied through a manifold system emanating from a vaporizer connected to a 125,000 $ft.^3$ capacity liquid N_2 receiver. To check, the quality of the N_2 being supplied to the test bearings, a Beckman Model 778 Oxygen Analyzer was used to perform the O_2 measurements.

The Model *778,* while not as sensitive as some oxygen analysis equipment, was considered to be satisfactory for this particular application. It is used to determine the combined O_2 content in the gas entering the tester and the gas escaping from the oil tank vent line. The O_2 content of the N_2 entering the tester is guaranteed by the supplier to be less than 5 ppm. Consequently, if the O_2 content of the N_2 at the vent line is identical to that entering the tester, it is reasonable to assume that an atmosphere containing not more than 5 ppm of O_2 is achieved in the tester.

The procedure used to check this was as follows. The Model 778 Range Selector is set on the most sensitive scale, which is I-5% O_2 full band width. At this range, accuracy is **4** 1% full scale at constant temperature. Linearity and repeatability are ± 0.5% and ± 1% full scale, respectively. The Analyzer probe is inserted into a small-neck reagent bottle along with a discharge line from the N_2 supply manifold. The bottle is capped with a top having a small hole, which reduces the possibility of an excessive pressure build-up in the bottle, The bottle is then purged **for** about 10 minutes, at which time the zero adjust is used to set the read-out dial on the Oxygen Analyzer at 0.5%. The reason for setting it at this level rather than 0 is merely a matter of convenience in that the .5% point is more accurately readable than the 0 point. The procedure is then repeated, using the Pine from the tester vent port. Since the exhaust gas contains some oil vapors, it is necessary to insert a condensator into the system. This is achieved by constructing a simple cold trap consisting of a copper coil immersed in a Dewar filled with acetone and dry ice. In the measurements made on the exhaust gas, the 0.5% level was achieved rapidly, indicating that the O_2 level of the escaping gas is essentially the same as that entering the tester.

The Oxygen Analyzer has also been used to obtain a measure of the absolute O_2 content, In this case, the zero oxygen level is set by immersing the probe into a solution of Na_2SO_3 and water, In each case, the vented gas caused the indicator to drive to the zero point indicating a low O_2 content. The overall sensitivity of the instrument was not sufficient to achieve a precise quantitative measurement of the O_2 level.

As with any other test equipment, some modifications were required after initial check-out tests, although in the present case these were held to a minimum and were related primarily to the oil temperature control, Initially, it was thought that external heating of both the tester body and test oil would be necessary to attain and maintain the 600F bearing temperature. However, during the check-out runs, the continual rise in temperature of the lubricating fluid indicated the need for cooling, partly due to the fact that the insulation of the lubricant tank was more effective than originally estimated. Hence, the power generated by the bearings was sufficient to maintain bearing and oil temperatures above 600F without external heating. In the check-out of the last test machine with XRM-177F, it was found that about 1/2 hour of operation would bring the oil temperature up from the 250F starting level to 600F.

Unfortunately, due to the high viscosity of all of the test oils at room temperature, it was necessary to heat them to 250F in order to achieve adequate pumpability. Consequently, in order to achieve control of the operating temperature, cooling coils were added to the salt bath,

A measurement of oil ΔT and flow rate showed that 10.6 HP was being removed by the oil from the bearing test housing and returned to the tank. A similar beat balance on the cooling water showed a removal of 6.2 HP. Therefore, 4.4 HP is being rejected from the oil tank and piping by convection. Since the bearing power loss was calculated at 13 HP, it can be inferred that 2.4 HP is rejected by convection of the bearing housing. The table below summarizes the power levels in the system.

1.	Bearing Power Loss (Calculated)	13.0 HP
2.	Bearing Housing Heat Rejection (By Difference)	2.4 HP
3.	Heat Removal by Oil (Measured)	10.6 HP
4.	Heat Rejection from Oil Tank (By Difference)	4.4 HP
5.	Heat Removal by Cooling Water (Measured)	6.2 HP

From this information it was determined that a cooling capacity between three and six HP would maintain the bearing outer race temperatures in the desired temperature range of 575F to 610F.

After experimenting with various lengths and diameters of cooling coils, it was determined that 25 feet of 3/8" bore Monel tubing would give the necessary heat transfer surface and would be short enough to prevent vaporization of the cooling water. Figure 16 shows that this system would control outer ring temperatures within the desired range. This cooling system was therefore, installed in the testers with automatic temperature sensors.

The decision made later in the program to adjust temperatures downward to 500F and 400F increased the need for additional heat exchanger capacity and consequently, required the investigation and fabrication of the proper size oil–water heat exchangers.

A second problem was that initially some foaming of both the MCS-354 and the XRM-177F was encountered. The addition of a small quantity (15 cc) of GE F–50 Silicone Fluid to the MCS-354 resulted in an almost immediate cessation of the foaming. The F–50, however, did not react similarly with the XRM-177F. In the ease of this fluid, 5 ml of a standard anti–foam agent (10% DC-200, 90% toluene) served to reduce, although not completely eliminate, foaming. To further decrease the foaming, several changes

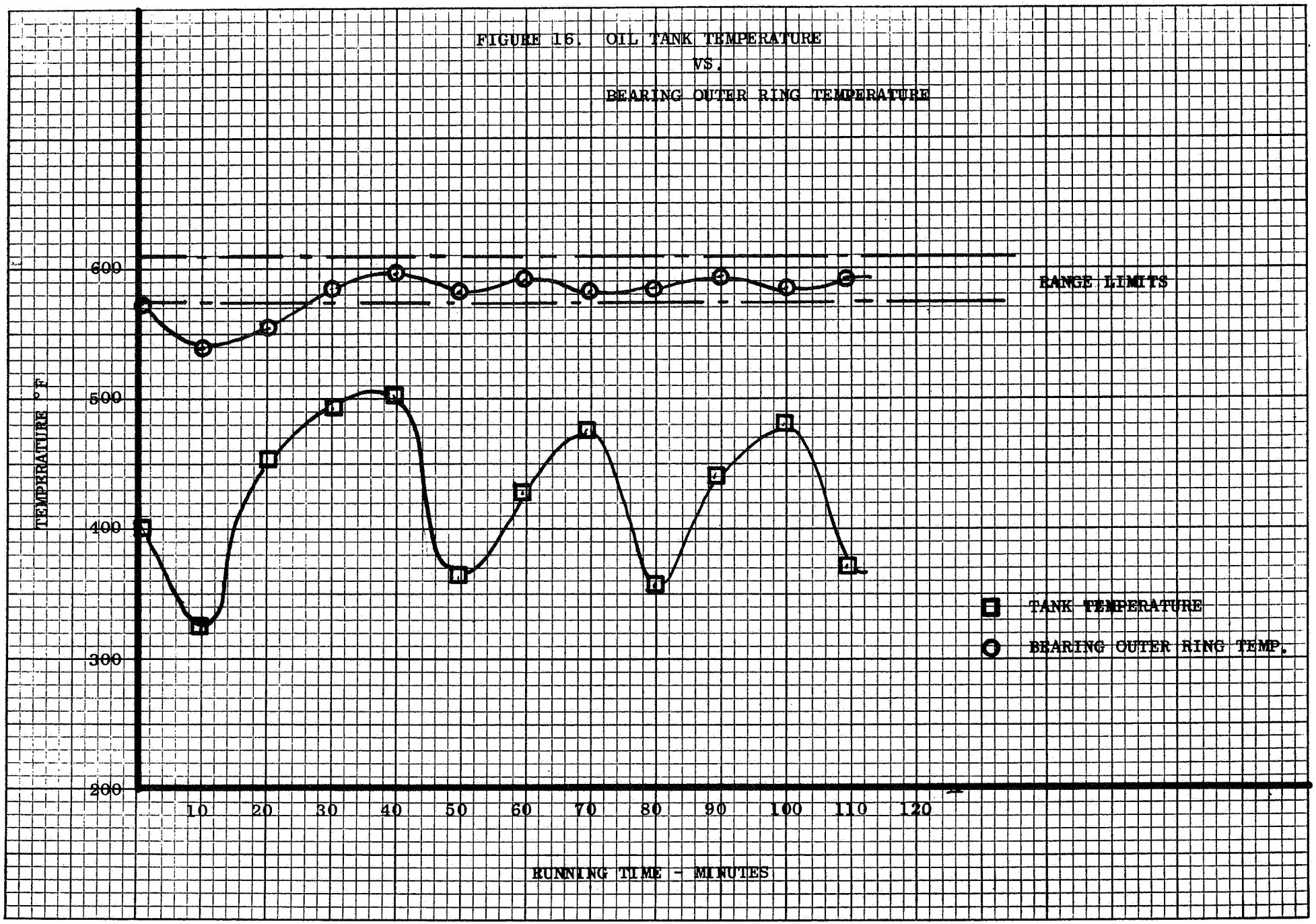


FIGURE 16. OIL TANK TEMPERATURE
VS.
BEARING OUTER RING TEMPERATURE
600
500
400
300
200
TEMPERATURE °F
10
20
30
40
50
60
70
80
90
100
110
120
RUNNING TIME - MINUTES
RANGE LIMITS
TANK TEMPERATURE
BEARING OUTER RING TEMP.

were made in the tester pump arrangement. The drain lines were shortened so that they now terminated above the level of the fluid. This eliminated the possibility of exposing the pump to a large amount of entrapped N_2, and allowed more settling time for foam. In addition, the oil flow to the center of the shaft was eliminated in order to reduce the total amount of oil in circulation, Lastly, the center drain line was closed up, forcing all of the oil to exit on the outside drains, thus assuring sufficient lubrication of the bearings.

There was some reservation regarding the last two modifications with regard to their effect on bearing temperature distribution. It was found, however, that temperature control was improved as evidenced by the fact that the outer races of the two test bearings now have the same temperature, where previously a 2° differential existed.

The addition of an anti-foaming agent to the mineral oil was discussed with Mobil Oil technical personnel, who expressed the opinion that the addition of this material would have no effect whatsoever on the performance of the XRM-177F. This information was reviewed with NASA personnel who had previously verbally approved the use of the anti-foam additive.

Other than the above two problems, no major difficulties were encountered with the testers. Some operational problems were encountered, however these were caused by the nature and behavior of the test fluids and will be discussed in the appropriate sections of this report.

5.0 Rolling Element Computer Analysis Program (RECAP)

The testing of precision bearings may appear to be a relatively simple task in that the bearings are placed in a suitable machine, a load is applied, the bearing is brought to speed and temperature and run until failure occurs. Unfortunately, this is a gross over-simplification. A rolling element bearing is a highly complex device and as such carries with it extremely complex stress patterns and variables affecting its ultimate performance.

The individual stress distributions, component kinematics and interactions must be known and controlled in order to achieve meaningful test data. The relationships of the rolling elements under actual engine operating conditions have always been a challenge to bearing designers due to a firm lack of understanding and because *of* the extremely tedious and relatively inaccurate hand calculations required to define these variables. Consequently, several years ago, Mr. A. B. Jones, under a consulting agreement with General Electric, developed a digital computer program to solve and clarify the above calculations.[6] This program now in general use by General Electric is known as RECAP (Rolling Element Computer Analysis Program). This program was used prior to ard during the testing phase to predict inner and outer race contact angles, stresses, and lives in an effort to eliminate stress differences and thus, fatigue life differences between test bearings due to contact angle and curvature variations.

An important input to the program were the results obtained on the RC Rig materials which indicated the relative degree of improvement and/or degradation which could be expected with the various fluids and at the proposed operating temperature.

A typical RECAP data sheet is shown in Figure 17. This particular set of conditions examines the bearing used in the current program at an axial load of 5800 lbs. and a speed of 12,000 rpm. As may be seen,

the fatigue life prediction for the entire bearing is ~15.3 x 10^6 revolutions which is ~21 hours. Multiplying this by five for M-50, it can be seen that the B_{10} life is close to the 100 hours B_{10} life set for this program. It should be pointed out that the AFBMA life prediction for this bearing is 18.7 hours using the race curvatures (f_1 = 54, f_0 = 52) used in the subject test bearings,

```
                    ROLLING ELEMENT COMPUTER ANALYSIS PROGRAM
                                    RECAP
                          GENERAL ELECTRIC COMPANY
                  AVIATION TECHNOLOGY AND DEMONSTRATOR PROGRAMS DEPARTMENT
                      ADVANCED BEARING AND SEAL TECHNOLOGY UNIT
CC MCCRE  X4407
NASA BRG

SYSTEM PARAMETERS

  BEARING     NUMBER OF    BALL/ROLL      PITCH        RACE CURVATURES        CONTACT     DIAMETRAL     INITIAL   DISTANCE TO
   NUMBER   BALLS/ROLLS     DIAMETER     DIAMETER      OUTER      INNER        ANGLE      CLEARANCE     PRELOAD  MOMENT CENTR
      1      1.5000E 01  8.1250E-01  6.1023E 00  5.2000E-01  5.4000E-01  1.0000E 01  -0.           -0.        1.8000E 01
      2      2.8000E 01  4.3300E-01  4.6800E 00  -0.         -0.                     5.0000E-03  -0.        -1.8000E 01

  BEARING      CONTROL    BALL/ROLL      ELASTIC MODULUS          POISSONS RATIO        FRICTION      TOTAL   1 =CALCULATE
   NUMBER   ASSUMPTION     DENSITY   BALLS/ROLLS     RACES   BALLS/ROLLS     RACES   COEFFICIENT  ROLL LENGTH   STRESSES
      1      1.0000E 00  2.8300E-01  2.9000E 07  2.2000E 07  2.5000E-01  2.5000E-01  7.5000E-02  -0.           1.0000E 00
      2      -0.         2.8300E-01  2.9000E 07  2.2000E 07  2.5000E-01  2.5000E-01  7.5000E-02  4.3000E-01   1.0000E 00

  BEARING    EFFECTIVE     SPIN/ROLL CONSTANT      SPIN/ROLL EXPONENT      FATIGUE CONSTANT        ROLL LAMBDA
   NUMBER   ROLL LENGTH     OUTER      INNER       OUTER      INNER      OUTER      INNER       OUTER      INNER
      1      -0.         -0.         -0.         -0.         -0.        7.1400E 03  7.1400E 03  0.         -0.
      2      3.0000E-02  -0.         -0.         -0.         -0.        4.9500E 04  4.9500E 04  0.1000E-01  0.100E-01

  BEARING      PHASE      CONSTANT   1 = FLOAT                    INITIAL DISPLACEMENTS                          SHIM
   NUMBER      ANGLE       THRUST    RADIALLY   ALONG X    ALONG Y    ALONG Z    ABOUT Y    ABOUT Z    OUT.    IN.
      1      -0.         -0.         -0.        -0.        -0.        -0.        -0.        -0.        0.      0.
      2      -0.         -0.         -0.        -0.        -0.        -0.        -0.        -0.        0.      0.

INPUT DATA FOR LOAD CONDITION  1

        EXTERNAL LOADS APPLIED AT MOMENT CENTER                    RPM OF      RPM OF      ROTATION FACTOR
  ALONG X     ALONG Y     ALONG Z     ABOUT Y     ABOUT Z        OUTER       INNER       OUTER      INNER
-5.8000E 03  -0.         2.0000E 00  -0.         -0.         -0.         1.2000E 04  0           1.0000E 00

LOAD VECTOR FIXED WITH RESPECT TO OUTER RACES

LOAD VECTOR ROTATES WITH RESPECT TO INNER RACES


  6 ITERATIONS WITH 5 BALL ID 5 ELEMENTS PLUS  1 WITH COMPLETE COMPLEMENT

  SYSTEM OUTPUT DATA

  BEARING                    REACTIONS OF BEARINGS ON SHAFT
   NUMBER      ALONG X     ALONG Y     ALONG Z     ABOUT Y     ABOUT Z
      1      5.8000E 03  -1.0071E-03  -1.9993E 00  3.6001E 01  -1.3657E-03
      2      0.          0.          0.          0.          0.

                  DISPLACEMENTS OF SHAFT AT MOMENT CENTER
              ALONG X     ALONG Y     ALONG Z     ABOUT Y     ABOUT Z
            4.7202E 00  -7.1615E-10  -4.7747E-04  2.4567E-05  5.3077E-11

                  NON-LINEAR SPRING RATES AT MOMENT CENTER
              ALONG X     ALONG Y     ALONG Z     ABOUT Y     ABOUT Z
            2.0642E 06  4.1225E 06  4.1226E 06  1.5551E 09  1.5551E 09
```

FIGURE 17

OUTPUT CONE BEARING NO. 1

RELATIVE DISPLACEMENTS AT BEARING CENTER

ALONG X	ALONG Y	ALONG Z	ABOUT Y	ABOUT Z
4.7202E-03	-5.7498E-11	-3.5262E-05	2.4567E-05	9.0265E-11

NON-LINEAR SPRING RATES AT BEARING CENTER

ALONG X	ALONG Y	ALONG Z	ABOUT Y	ABOUT Z
2.0642E 06	4.1225E 06	4.1226E 06	9.8018E 06	9.8017E 06

FATIGUE LIFE

OUTER RACE	INNER RACE	BEARING	FRICTION TORQUE
1.2542E 02	1.6675E 01	1.5227E 01	4.2429E 01

BALL/ROLL NUMBER	ANGULAR POSITION	BALL/ROLL LOADS OUTER	BALL/ROLL LOADS INNER	OPERATING CONTACT ANGLES OUTER	OPERATING CONTACT ANGLES INNER	CENTRIFUGAL FORCE	GYROSCOPIC MOMENT	TYPE OF CONTROL	VECTOR ATTITUDE
1	0.	1.0209E 03	8.4511E 02	2.0797E 01	2.7348E 01	1.9395E 02	-1.1191E 01	000000	1.8390E 01
2	2.4000E 01	1.0209E 03	8.4506E 02	2.0790E 01	2.7339E 01	1.9395E 02	-1.1187E 01	000000	1.8384E 01
3	4.8000E 01	1.0208E 03	8.4493E 02	2.0771E 01	2.7314E 01	1.9394E 02	-1.1176E 01	000000	1.8367E 01
4	7.2000E 01	1.0206E 03	8.4474E 02	2.0742E 01	2.7277E 01	1.9391E 02	-1.1160E 01	000000	1.8341E 01
5	9.6000E 01	1.0204E 03	8.4454E 02	2.0709E 01	2.7235E 01	1.9389E 02	-1.1142E 01	000000	1.8312E 01
6	1.2000E 02	1.0203E 03	8.4436E 02	2.0677E 01	2.7194E 01	1.9387E 02	-1.1124E 01	000000	1.8284E 01
7	1.4400E 02	1.0202E 03	8.4424E 02	2.0652E 01	2.7163E 01	1.9385E 02	-1.1110E 01	000000	1.8262E 01
8	1.6800E 02	1.0201E 03	8.4417E 02	2.0639E 01	2.7145E 01	1.9384E 02	-1.1103E 01	000000	1.8250E 01
9	1.9200E 02	1.0201E 03	8.4417E 02	2.0639E 01	2.7145E 01	1.9384E 02	-1.1103E 01	000000	1.8250E 01
10	2.1600E 02	1.0202E 03	8.4424E 02	2.0652E 01	2.7163E 01	1.9385E 02	-1.1110E 01	000000	1.8262E 01
11	2.4000E 02	1.0203E 03	8.4436E 02	2.0677E 01	2.7194E 01	1.9387E 02	-1.1124E 01	000000	1.8284E 01
12	2.6400E 02	1.0204E 03	8.4454E 02	2.0709E 01	2.7235E 01	1.9389E 02	-1.1142E 01	000000	1.8312E 01
13	2.8800E 02	1.0206E 03	8.4474E 02	2.0742E 01	2.7277E 01	1.9391E 02	-1.1160E 01	000000	1.8341E 01
14	3.1200E 02	1.0208E 03	8.4493E 02	2.0771E 01	2.7314E 01	1.9394E 02	-1.1176E 01	000000	1.8367E 01
15	3.3600E 02	1.0209E 03	8.4506E 02	2.0790E 01	2.7339E 01	1.9395E 02	-1.1187E 01	000000	1.8384E 01

BALL/ROLL NUMBER	GYRO SLIP COEFFICIENT	ORBITAL VELOCITY	ROTATIONAL VELOCITY	SPINNING VELOCITY	ROLLING VELOCITY OUTER	ROLLING VELOCITY INNER	SPIN/ROLL RATIO	SPINNING TORQUES OUTER	SPINNING TORQUES INNER
1	2.6982E-02	5.3079E 03	-4.4877E 04	1.0062E 04	-3.9875E 04	5.0274E 04	2.0014E-01	3.4608E 00	2.1155E 00
2	2.6974E-02	5.3078E 03	-4.4877E 04	1.0059E 04	-3.9875E 04	5.0274E 04	2.0007E-01	3.4606E 00	2.1153E 00
3	2.6950E-02	5.3077E 03	-4.4876E 04	1.0050E 04	-3.9874E 04	5.0276E 04	1.9990E-01	3.4601E 00	2.1149E 00
4	2.6916E-02	5.3074E 03	-4.4874E 04	1.0038E 04	-3.9871E 04	5.0278E 04	1.9964E-01	3.4594E 00	2.1142E 00
5	2.6876E-02	5.3071E 03	-4.4873E 04	1.0023E 04	-3.9869E 04	5.0280E 04	1.9934E-01	3.4586E 00	2.1136E 00
6	2.6838E-02	5.3068E 03	-4.4871E 04	1.0009E 04	-3.9867E 04	5.0283E 04	1.9906E-01	3.4580E 00	2.1130E 00
7	2.6808E-02	5.3065E 03	-4.4870E 04	9.9981E 03	-3.9865E 04	5.0285E 04	1.9883E-01	3.4575E 00	2.1126E 00
8	2.6791E-02	5.3064E 03	-4.4869E 04	9.9921E 03	-3.9864E 04	5.0286E 04	1.9871E-01	3.4573E 00	2.1123E 00
9	2.6791E-02	5.3064E 03	-4.4869E 04	9.9921E 03	-3.9864E 04	5.0286E 04	1.9871E-01	3.4573E 00	2.1123E 00
10	2.6808E-02	5.3065E 03	-4.4870E 04	9.9981E 03	-3.9865E 04	5.0285E 04	1.9883E-01	3.4575E 00	2.1126E 00
11	2.6838E-02	5.3068E 03	-4.4871E 04	1.0009E 04	-3.9867E 04	5.0283E 04	1.9906E-01	3.4580E 00	2.1130E 00
12	2.6876E-02	5.3071E 03	-4.4873E 04	1.0023E 04	-3.9869E 04	5.0280E 04	1.9934E-01	3.4586E 00	2.1136E 00
13	2.6916E-02	5.3074E 03	-4.4874E 04	1.0038E 04	-3.9871E 04	5.0278E 04	1.9964E-01	3.4594E 00	2.1142E 00
14	2.6950E-02	5.3077E 03	-4.4876E 04	1.0050E 04	-3.9874E 04	5.0276E 04	1.9990E-01	3.4601E 00	2.1149E 00
15	2.6974E-02	5.3078E 03	-4.4877E 04	1.0059E 04	-3.9875E 04	5.0274E 04	2.0007E-01	3.4606E 00	2.1153E 00

BALL/ROLL NUMBER	ROLLING TORQUES OUTER	ROLLING TORQUES INNER	OPERATING PITCH DIA.	SEMI-MAJOR AXIS OUTER	SEMI-MAJOR AXIS INNER	SEMI-MINOR AXIS OUTER	SEMI-MINOR AXIS INNER	MEAN COMPRESSIVE STRESS OUTER	MEAN COMPRESSIVE STRESS INNER
1	2.0683E-01	8.9745E-02	6.1039E 00	1.1753E-01	8.5674E-02	1.5518E-02	1.4586E-02	1.7817E 05	2.1527E 05
2	2.0682E-01	8.9737E-02	6.1039E 00	1.1753E-01	8.5673E-02	1.5518E-02	1.4585E-02	1.7817E 05	2.1527E 05
3	2.0678E-01	8.9714E-02	6.1039E 00	1.1753E-01	8.5668E-02	1.5518E-02	1.4584E-02	1.7816E 05	2.1526E 05
4	2.0673E-01	8.9681E-02	6.1039E 00	1.1752E-01	8.5662E-02	1.5517E-02	1.4583E-02	1.7815E 05	2.1525E 05
5	2.0667E-01	8.9647E-02	6.1039E 00	1.1751E-01	8.5656E-02	1.5516E-02	1.4581E-02	1.7814E 05	2.1524E 05
6	2.0662E-01	8.9616E-02	6.1039E 00	1.1751E-01	8.5650E-02	1.5516E-02	1.4580E-02	1.7813E 05	2.1523E 05
7	2.0658E-01	8.9594E-02	6.1039E 00	1.1750E-01	8.5646E-02	1.5515E-02	1.4579E-02	1.7812E 05	2.1522E 05
8	2.0657E-01	8.9583E-02	6.1039E 00	1.1750E-01	8.5644E-02	1.5515E-02	1.4578E-02	1.7812E 05	2.1522E 05
9	2.0657E-01	8.9583E-02	6.1039E 00	1.1750E-01	8.5644E-02	1.5515E-02	1.4578E-02	1.7812E 05	2.1522E 05
10	2.0658E-01	8.9594E-02	6.1039E 00	1.1750E-01	8.5646E-02	1.5515E-02	1.4579E-02	1.7812E 05	2.1522E 05
11	2.0662E-01	8.9616E-02	6.1039E 00	1.1751E-01	8.5650E-02	1.5516E-02	1.4580E-02	1.7813E 05	2.1523E 05
12	2.0667E-01	8.9647E-02	6.1039E 00	1.1751E-01	8.5656E-02	1.5516E-02	1.4581E-02	1.7814E 05	2.1524E 05
13	2.0673E-01	8.9681E-02	6.1039E 00	1.1752E-01	8.5662E-02	1.5517E-02	1.4583E-02	1.7815E 05	2.1525E 05
14	2.0678E-01	8.9714E-02	6.1039E 00	1.1753E-01	8.5668E-02	1.5518E-02	1.4584E-02	1.7816E 05	2.1526E 05
15	2.0682E-01	8.9737E-02	6.1039E 00	1.1753E-01	8.5673E-02	1.5518E-02	1.4585E-02	1.7817E 05	2.1527E 05

FIGURE 17 (CONTD)

BALL/ROLL NUMBER	UPPER EDGE OF PATH OUTER	UPPER EDGE OF PATH INNER	LOWER EDGE OF PATH OUTER	LOWER EDGE OF PATH INNER	H/D VALUES OUTER	H/D VALUES INNER	SV VALUE
1	3.6949E 01	3.8608E 01	4.6453E 00	1.6087E 01	1.0443E-01	1.1803E-01	1.2145E 00
2	3.6942E 01	3.8599E 01	4.6386E 00	1.6079E 01	1.0439E-01	1.1797E-01	1.2141E 00
3	3.6921E 01	3.8573E 01	4.6196E 00	1.6054E 01	1.0428E-01	1.1782E-01	1.2130E 00
4	3.6892E 01	3.8536E 01	4.5917E 00	1.6018E 01	1.0412E-01	1.1760E-01	1.2113E 00
5	3.6858E 01	3.8493E 01	4.5596E 00	1.5977E 01	1.0393E-01	1.1735E-01	1.2094E 00
6	3.6825E 01	3.8452E 01	4.5288E 00	1.5937E <1	1.0376E-01	1.1711E-01	1.2076E 00
7	3.6800E 01	3.8419E 01	4.5048E 00	1.5906E 01	1.0362E-01	1.1692E-01	1.2062E 00
8	3.6786E 01	3.8402E 01	4.4917E 00	1.5889E 01	1.0355E-01	1.1682E-01	1.2054E 00
9	3.6786E 01	3.8402E 01	4.4917E 00	1.5889E <1	1.0355E-01	1.1682E-01	1.2054E 00
10	3.6800E 01	3.8419E 01	4.5048E 00	1.5906E 01	1.0362E-01	1.1692E-01	1.2062E 00
11	3.6825E 01	3.8452E 01	4.5288E 00	1.5937E 01	1.0376E-01	1.1711E-01	1.2076E 00
12	3.6858E 01	3.8493E 01	4.5596E 00	1.5977E 01	1.0393E-01	1.1735E-01	1.2094E 00
13	3.6892E 01	3.8536E 01	4.5917E 00	1.6018E <1	1.0412E-01	1.1760E-01	1.2113E 00
14	3.6921E 01	3.8573E 01	4.6196E 00	1.6054E 01	1.0428E-01	1.1782E-01	1.2130E 00
15	3.6942E 01	3.8599E 01	4.6386E 00	1.6079E 01	1.0439E-01	1.1797E-01	1.2141E 00

BALL/ROLL NUMBER	MAXIMUM ORTHOGONAL SHEAR STRESS OUTER PSI	OUTER DEPTH	INNER PSI	INNER DEPTH	MAXIMUM SHEAR STRESS OUTER PSI	OUTER DEPTH	INNER PSI	INNER DEPTH
1	1.2544E 02	7.6611E-03	1.0382E 02	7.1428E-03	-8.5970E 04	1.1777E-02	-1.0501E 05	1.0378E-02
2	1.2543E 02	7.6610E-03	1.0382E 02	7.1427E-03	-8.5969E 04	1.1777E-02	-1.0501E 05	1.0378E-02
3	1.2542E 02	7.6608E-03	1.0380E 02	7.1422E-03	-8.5965E 04	1.1776E-02	-1.0500E 05	1.0377E-02
4	1.2540E 02	7.6604E-03	1.0378E 02	7.1415E-03	-8.5960E 04	1.1776E-02	-1.0500E 05	1.0376E-02
5	1.2538E 02	7.6601E-03	1.0375E 02	7.1407E-03	-8.5955E 04	1.1775E-02	-1.0499E 05	1.0375E-02
6	1.2536E 02	7.6599E-03	1.0373E 02	7.1401E-03	-8.5949E 04	1.1775E-02	-1.0498E 05	1.0374E-02
7	1.2535E 02	7.6597E-03	1.0371E 02	7.1396E-03	-8.5946E 04	1.1775E-02	-1.0498E 05	1.0373E-02
8	1.2534E 02	7.6596E-03	1.0371E 02	7.1393E-<3	-8.5944E 04	1.1775E-02	-1.0498E 05	1.0373E-02
9	1.2534E 02	7.6596E-03	1.0371E 02	7.1393E-03	-8.5944E 04	1.1775E-02	-1.0498E 05	1.0373E-02
10	1.2535E 02	7.6597E-03	1.0371E 02	7.1396E-03	-8.5946E 04	1.1775E-02	-1.0498E 05	1.0373E-02
11	1.2536E 02	7.6599E-03	1.0373E 02	7.1401E-03	-8.5949E 04	1.1775E-02	-1.0498E 05	1.0374E-02
12	1.2538E 02	7.6601E-03	1.0375E 02	7.1407E-03	-8.5955E 04	1.1775E-02	-1.0499E 05	1.0375E-02
13	1.2540E 02	7.6604E-03	1.0378E 02	7.1415E-<3	-8.5960E 04	1.1776E-02	-1.0500E 05	1.0376E-02
14	1.2542E 02	7.6608E-03	1.0380E 02	7.1422E-03	-8.5965E 04	1.1776E-02	-1.0500E 05	1.0377E-02
15	1.2543E 02	7.6610E-03	1.0382E 02	7.1427E-03	-8.5969E 04	1.1777E-02	-1.0501E 05	1.0378E-02

OUTPUT FOR BEARING NO 2

BEARING NO 2 IS COMPLETELY UNLOADED

FIGURE 17 (CONTD)

6.0 Test Results

6.1 RC Rig Tests

RC Rig testing was performed under the following conditions for all but one test series.

Temperature:	Metal 600F ± 5F
	Oil 500F ± 20F
Load:	700,000 psi max. hertz
Speed:	25,000 stress cycles/min.
	(1350 fpm surface speed)
Atmosphere:	Air

One test series was performed in an inerted atmosphere to determine whether this would have any effect on the test results. The results of this test will be discussed later,

The first series of RC Rig tests was conducted on the three base fluids and CVM M-50 material. The data from these tests are presented in Weibull distribution in Figures 18 through 20. These data as well as the composite curve (Figure 21) have been published in an NASA TN dated May, 1967.(7) The three curves in the above figures are a composite of essentially three heats of M-50. A master heat (Latrobe 54265) used by General Electric for rig qualification and verification studies and the two CVM billet heats used respectively for the rings and rolling elements of the test bearings. The data, taken as a composite show the following trend:

Lubricant	B_{10} Life (Millions of Stress) Cycles	B_{50} Life (Millions of Stress) Cycles	Weibull Slope
Mobil XRM-177F	2.35	7.26	1.7
Monsanto MCS-354	.69	2.70	1.4
DuPont PR-143	1.04	6.76	1.0
AFBMA *	1.4	-	-

* The AFBMA value is shown for the RC Rig at a maximum hertz stress of 700,000 psi and for CVM M-50.

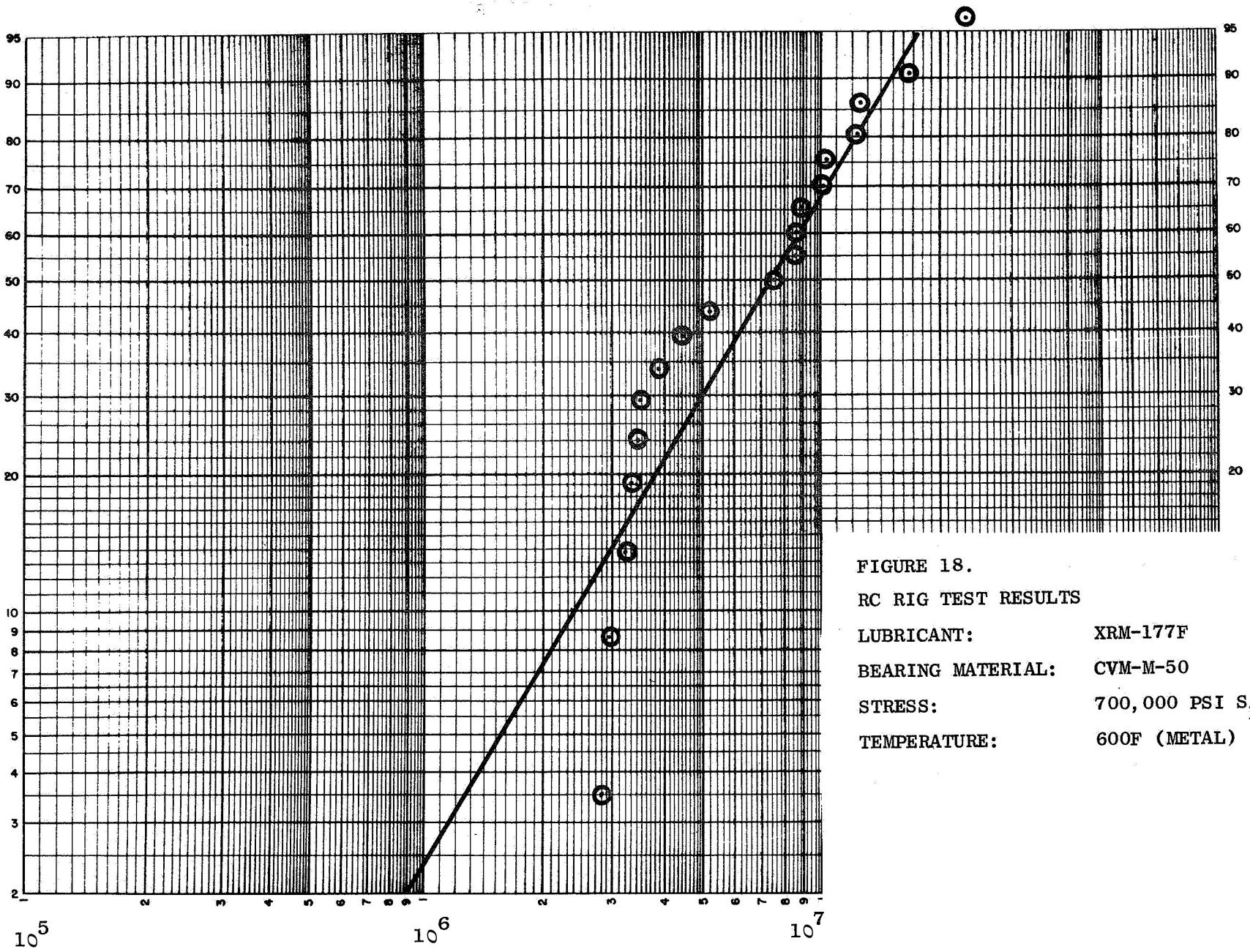


FIGURE 18.

RC RIG TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	CVM-M-50
STRESS:	700,000 PSI S_{MAX}
TEMPERATURE:	600F (METAL)

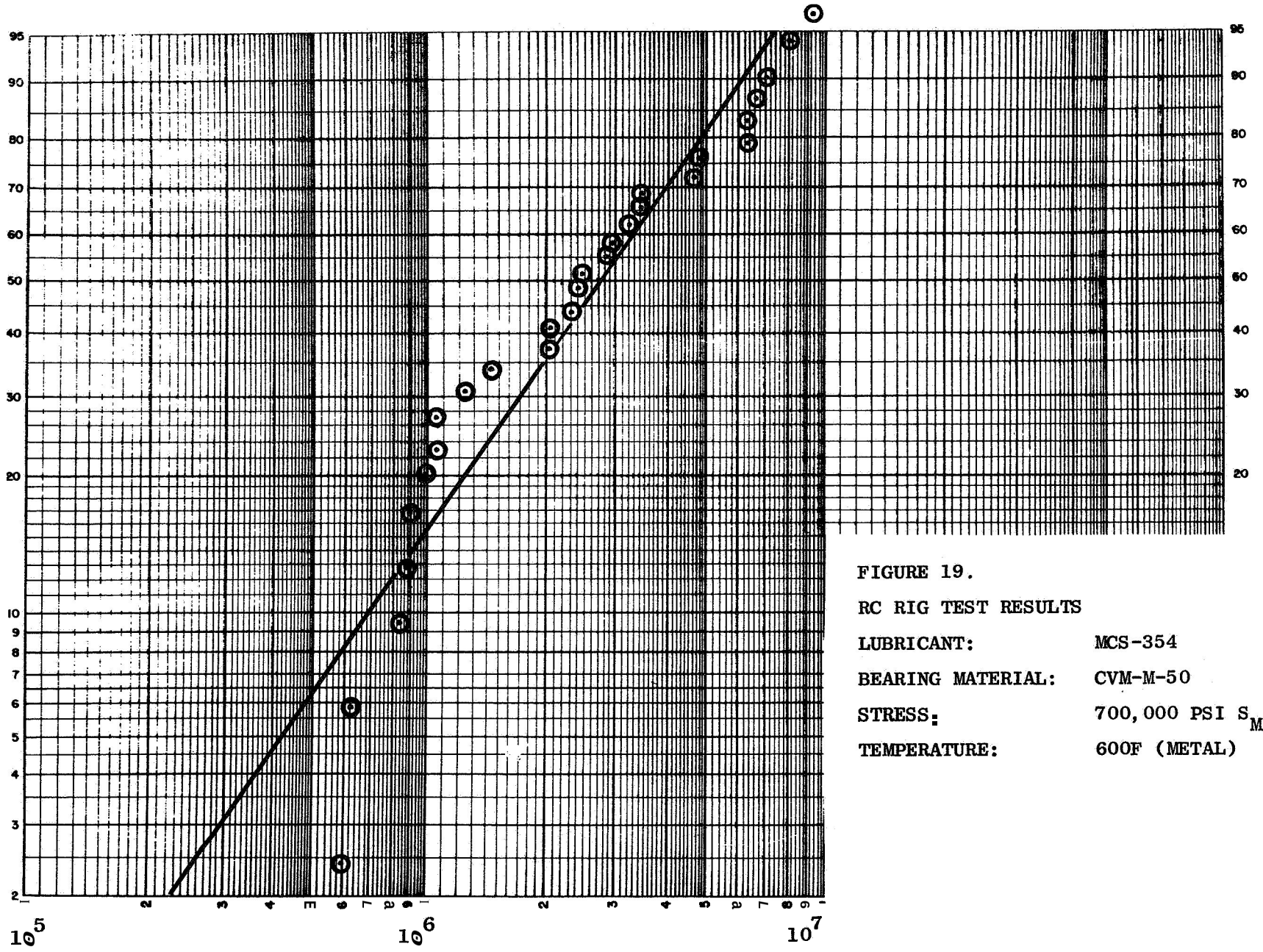


FIGURE 19.

RC RIG TEST RESULTS

LUBRICANT:	MCS-354
BEARING MATERIAL:	CVM-M-50
STRESS:	700,000 PSI S_{MAX}
TEMPERATURE:	600F (METAL)

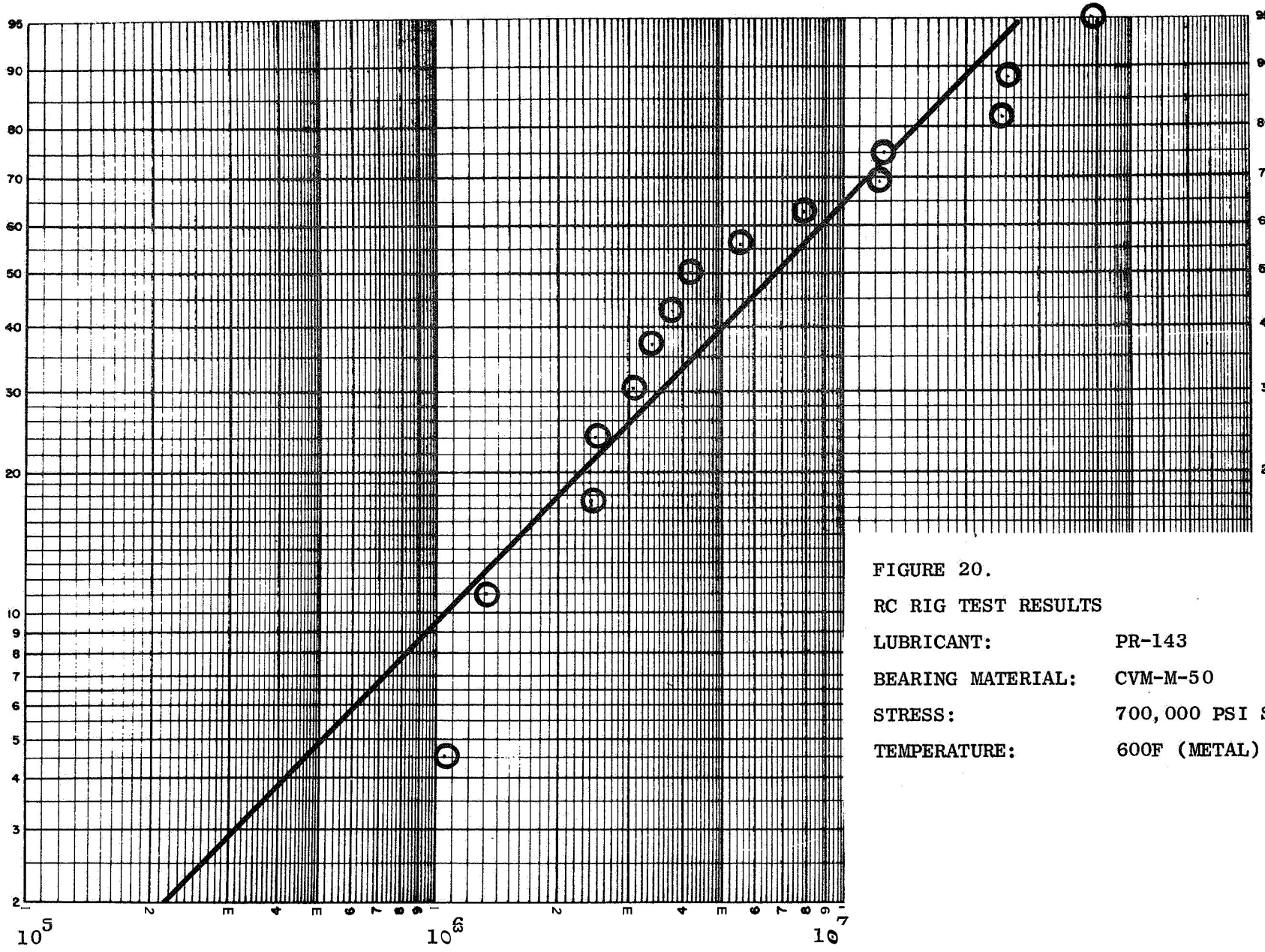


FIGURE 20.

RC RIG TEST RESULTS

LUBRICANT:	PR-143
BEARING MATERIAL:	CVM-M-50
STRESS:	700,000 PSI S_{MAX}
TEMPERATURE:	600F (METAL)

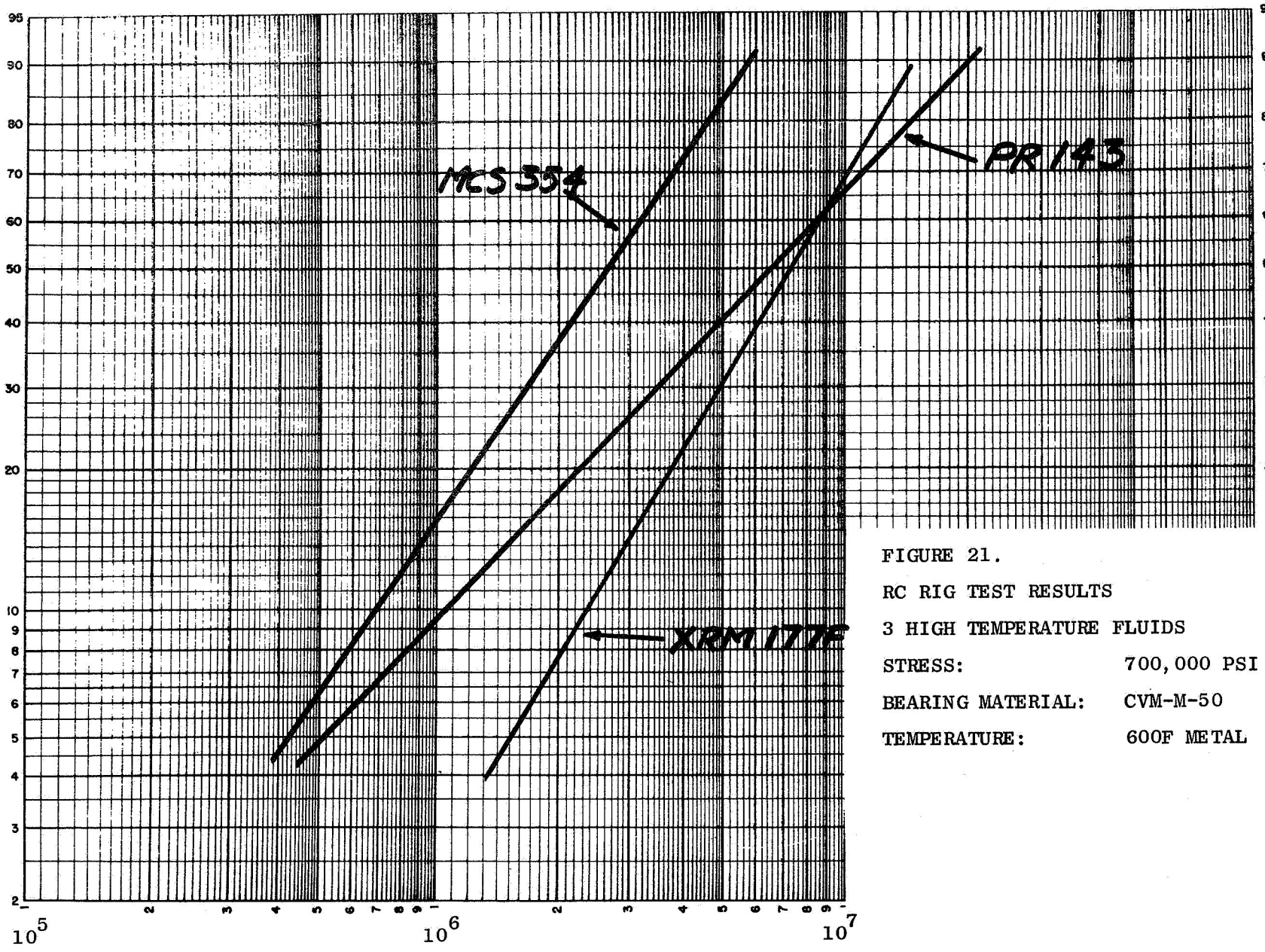


FIGURE 21.

RC RIG TEST RESULTS

3 HIGH TEMPERATURE FLUIDS

STRESS: 700,000 PSI S_{MAX}

BEARING MATERIAL: CVM-M-50

TEMPERATURE: 600F METAL

The above tabulation shows that the DuPont material will provide approximately AFBMA life, the XRM-177F should better this by about a factor of two, whereas the MCS-354 will show a drastic reduction in fatigue life.

These rankings were verified by the full scale bearing tests as will be shown later.

Secondly, it is interesting to individually examine in the RC Rig the two CVM billet heats of M-50 used for ring and ball manufacture. This is illustrated for XRM-177F in Figure 22 and here it is shown that the ball material is lower in fatigue life versus the ring material (B_{10} of 2,9 versus B_{10} of 6.8). The same trend is shown in Figure 23 which compares these two billet heats tested with the Monsanto MCS-354, These observations led to the conclusion that a greater incidence of ball failures should be expected than one would normally expect on a statistical basis. Again, as will be shown later, this was indeed the case. It should be pointed out that this type of prior information is extremely valuable, as armed with this knowledge, the tendency to blame the test or test conditions for the unusual number of ball failures is eliminated and along with it much needless, time consuming and expensive re-examination of test procedures, etc.

The second major test series was conducted with the most promising of the three test fluids (XRM-177F) and the WB-49 bearing material. This data is presented in Weibull distribution in Figure 24. The following life comparison can be made:

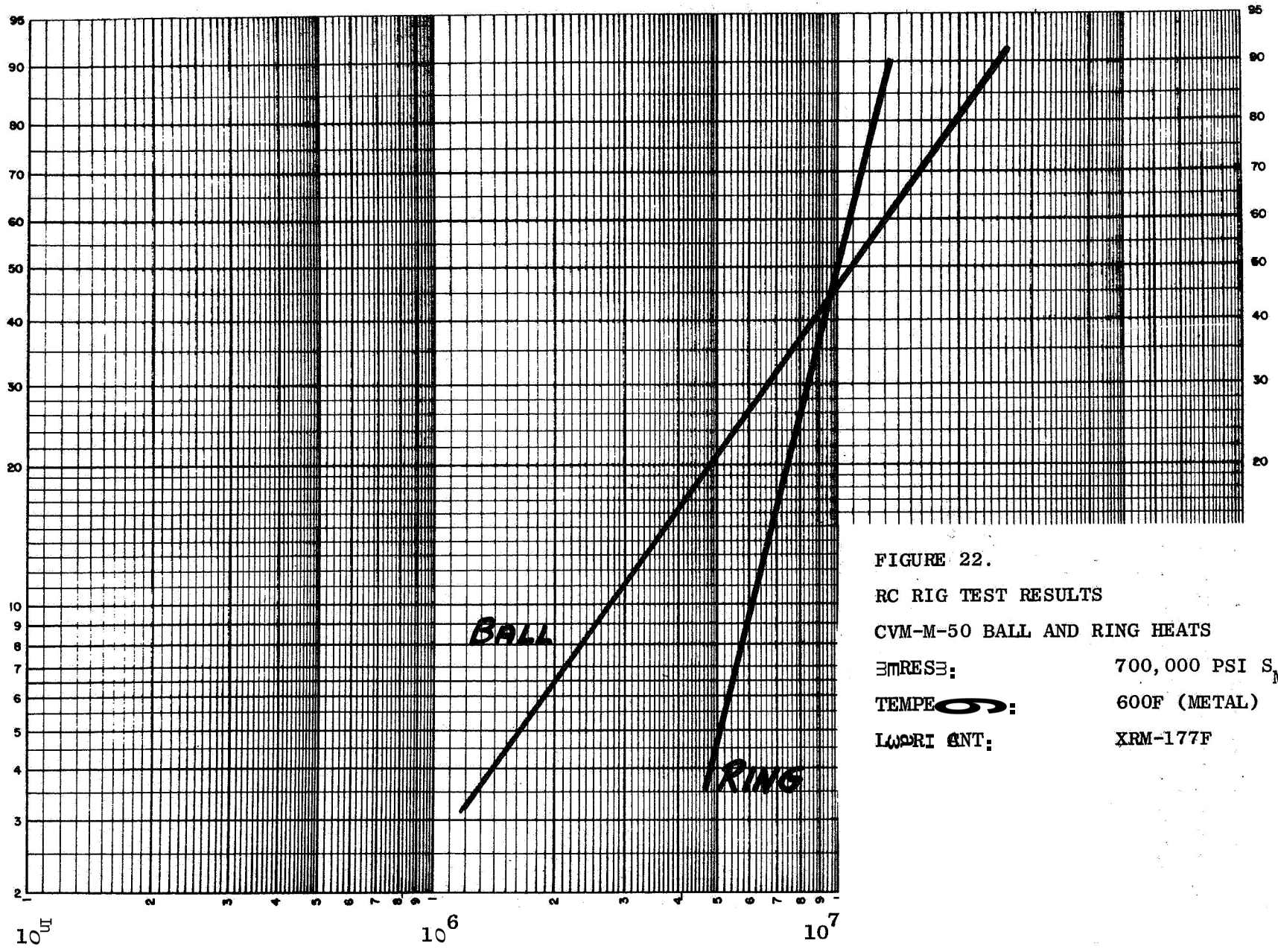


FIGURE 22.

RC RIG TEST RESULTS

CVM-M-50 BALL AND RING HEATS

STRESS: 700,000 PSI S_{MAX}

TEMPE[illegible]: 600F (METAL)

LUBRICANT: XRM-177F

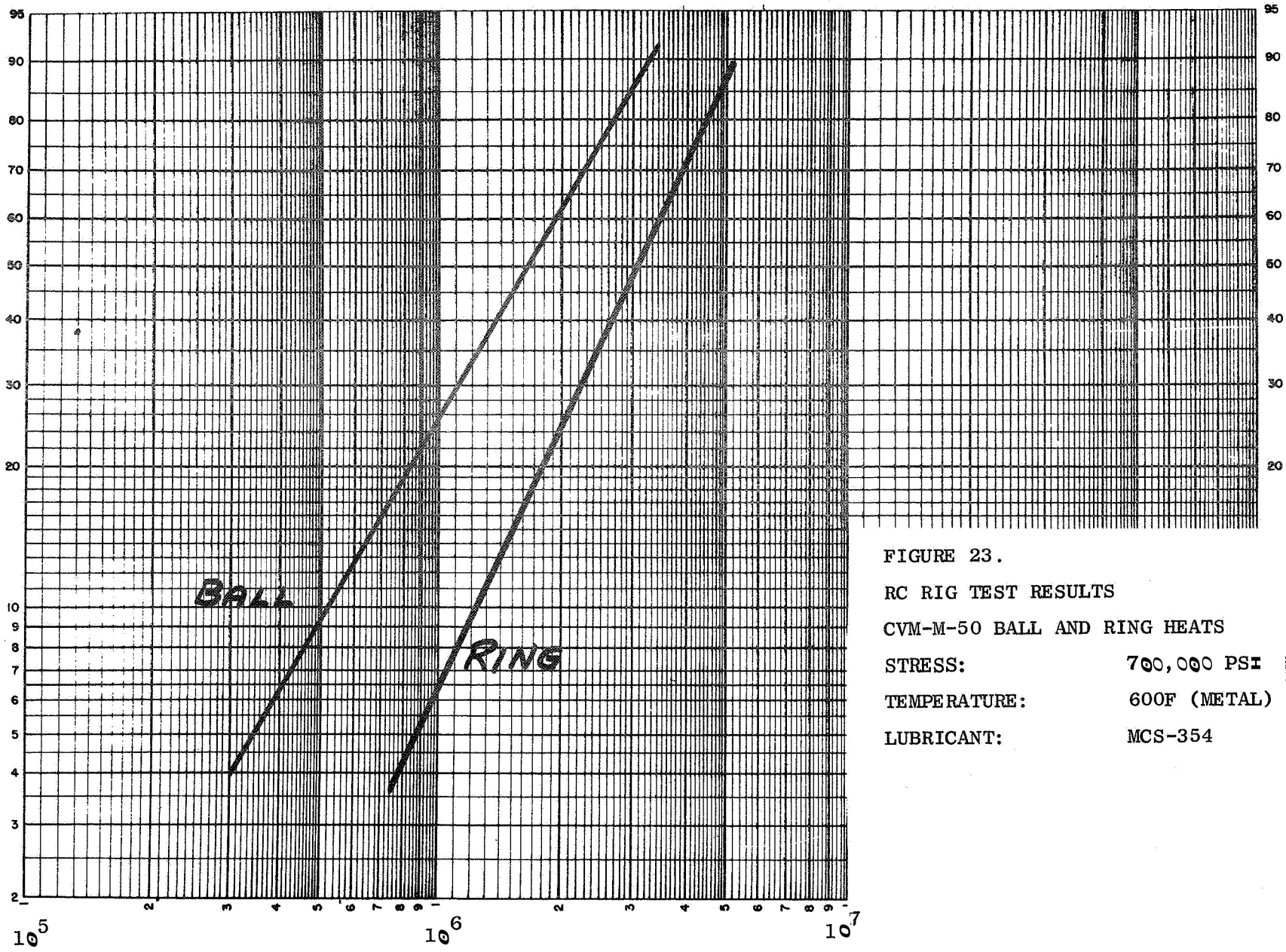


FIGURE 23.

RC RIG TEST RESULTS

CVM-M-50 BALL AND RING HEATS

STRESS: 700,000 PSI $\equiv$ MAX

TEMPERATURE: 600F (METAL)

LUBRICANT: MCS-354

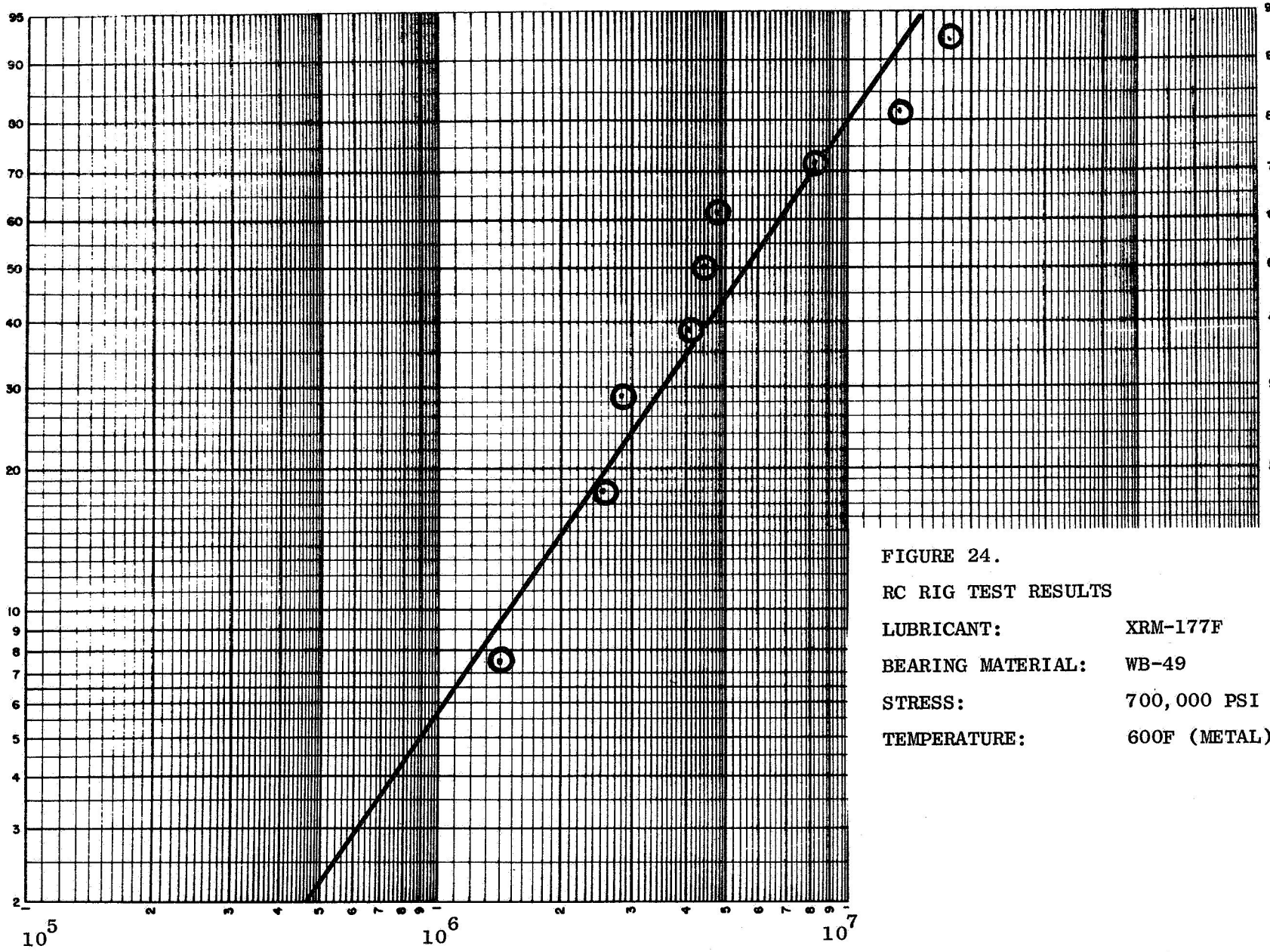


FIGURE 24.

RC RIG TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	WB-49
STRESS:	700,000 PSI S_{MAX}
TEMPERATURE:	600F (METAL)

Test Material	Fluid	B_{10} Life (Stress Cycle)	B_{50} Life (Stress Cycle)
M-50	XRM-177F	3.85×10^6	9.0×10^6
WB-49	XRM-177F	1.45×10^6	5.6×10^6

These data indicate the WB-49 to be inferior to the CVM M-50, Again, this observation based on RC Rig testing was verified in subsequent full scale tests.

The last test series which was performed was a determination of the effect of inerting the test fluid, in this case the XRM-177F. As is shown in the curve in Figure 25 essentially no difference in fatigue life is observed when testing the XRM-177F in a nitrogen atmosphere or in air, This is not unexpected as the RC Rig is a once-through system and test times are relatively short, ranging from <1 hour to ~8 hours, Consequently, any oxidation degradation of the oil which could affect the lubricant rheological properties would not have a significant effect on fatigue life. This may not be the case in the actual bearing tests, where the test fluid is recirculated and exposed to high temperatures for time periods as long as 750 hours. Here, inertion at least for two of the experimental fluids is essential to prevent fluid degradation and seizure of the operating components

6.2 Full Scale Bearing Tests

The full scale bearing test program as originally proposed consisted of the following schedule:

Test Series	Lubricant	Bearing Material	Test Temp.	B_{10}=	B_{50} Cut-Off	No. of Tests
I	XRM-177F	CVM M-50	600F	100 hr.	500 hr.	30
II	MCS-354	CVM M-50	600F	100 hr.	500 hr.	30
III	PR-143	CVM M-50	600F	100 hr,	500 hr,	30
IV	(Best of 3 above)	WB-49	600F	100 hr.	500 hr.	30
V	Same as IV	Best of 2 above	600F	50 hr.	250 hr.	30
VI	Same as IV	Same as V	600F	150 hr.	750 hr.	30

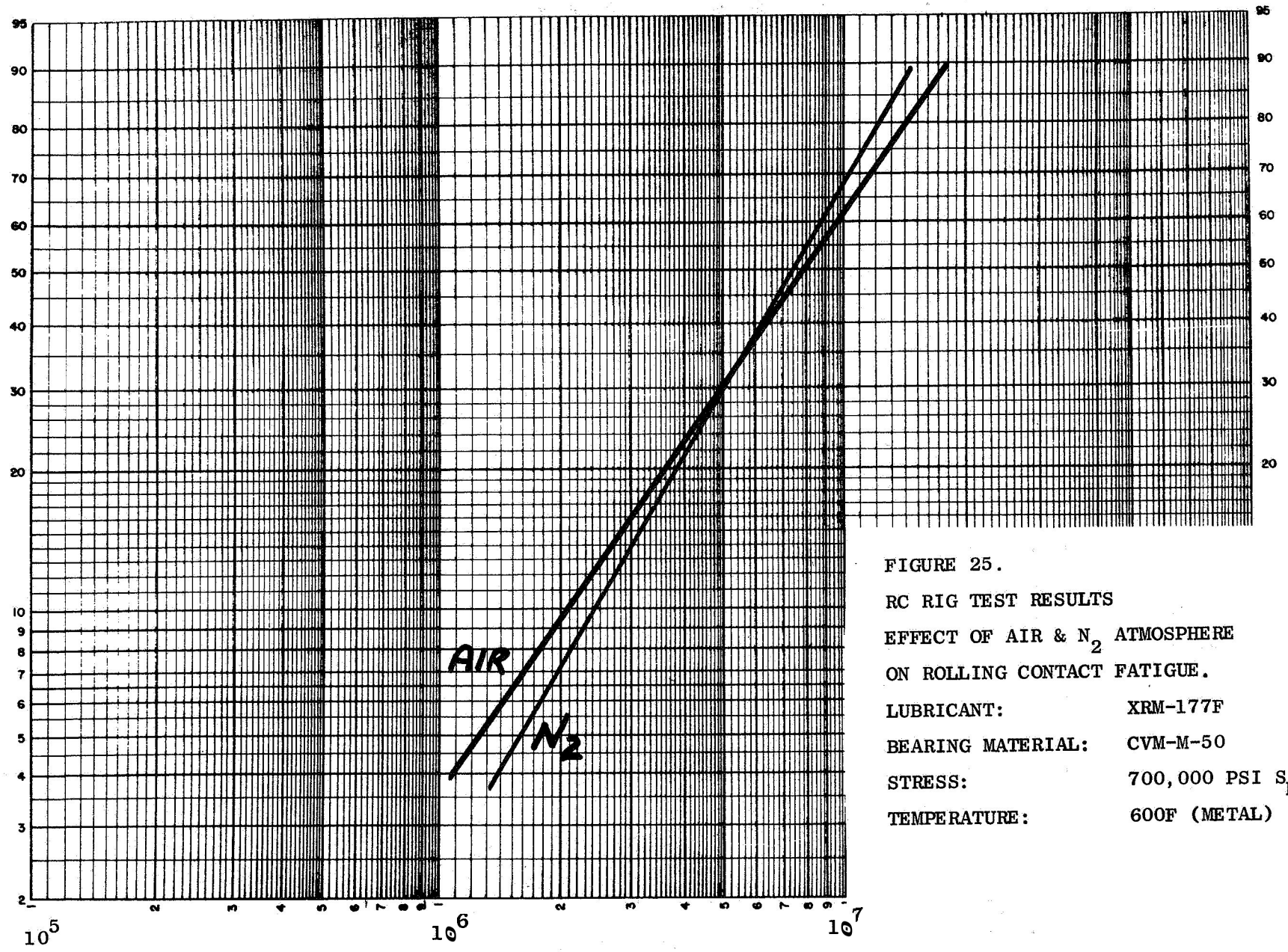


FIGURE 25.

RC RIG TEST RESULTS

EFFECT OF AIR & N_2 ATMOSPHERE ON ROLLING CONTACT FATIGUE.

LUBRICANT:	XRM-177F
BEARING MATERIAL:	CVM-M-50
STRESS:	700,000 PSI S_{MAX}
TEMPERATURE:	600F (METAL)

The logic behind this test sequence is obvious and has been dealt with in a previous section of this report. However, several events occurred during the course of the program which made it necessary to deviate from the original plan, Early in the program, it was observed that at the 5800 lb. axial load level, the XRM-177F at 600F was still maintaining a hydrodynamic film. In order to decrease the B_{10} life to 50 hours, or about one-half, the Hertzian load on the bearing would have to be increased by about 10%, assuming that the ninth power relationship between load and life was valid at 600F. It was felt that this would precipitate a mixed or boundary lubrication condition which could seriously affect the validity of the test results. To determine this, four 600F tests were conducted at a 7000 lb. axial load, As suspected, ball wear and some raceway surface distress was observed, indicating a breakdown of the hydrodynamic film with resultant asperity contact. In view of this, the program was re-oriented toward establishing the life-temperature relationship rather than the life-load relationship originally planned.

The second factor influencing the conduct of the program was the late delivery of the WB-49 bearings. The original plan had been to evaluate the best fluid determined in the first three test series with CVM M-50 bearings, with the WB-49 bearings, and then to use the best fluid-bearing material combination for the life-temperature series, The delay in obtaining the WB-49 bearings was so great however, that *a* decision was made to proceed with the life-temperature series using CVM M-50 bearings, The decision was made based on the relatively poor behavior of the WB-49 in the RC Rig tests, which indicated this material to be inferior to CVM M-50. The subsequent WB-49 bearing tests confirmed the RC Rig data and consequently the decision to proceed with the CVM M-50 was proven technically sound.

In order to maintain a logical sequence in detailing the test results, the WB-49 test series will be reported in their original position in the test schedule, Consequently, the actual test sequence and operating

conditions were as follows and will be discussed in this order.

Test Series	Lubricant	Bearing Material	Test Temp.	B_{10}=	B_{50} Cut-Off	No. of Tests
1	XRM-177F	CVM M-50	600F	100 hr.	500 hr.	30
II	Mcs-354	CVM M-50	600F	100 hr,	500 hr.	30
III	PR-143	CVM M-50	600F	100 hr.	500 hr.	30
IV	XRM-177F	WB-49	600F	100 hr.	500 hr.	30
V	XRM-177F	CVM M-50	500F	100 hr.	750 hr.(1)	$_{25}$(2)
VI	XRM-177F	CVM M-50	400F	100 hr,	750 hr,(1)	$_{25}$(2)

(1) In view of the decrease in temperature with the subsequent increase in lubricant viscosity, it was decided to increase the cut-off point to 750 hr. in order to obtain a sufficient number of failures to' produce a statistically valid population.

(2) The total number of tests were reduced because of the fact that several bearings had been used for exploratory tests earlier in the program and consequently a 30 bearing complement was not available.

One of the measures taken to insure test repeatability was the use of a new batch of oil with each new set of bearings, with the exception of Test Series VI, where oil which had been previously used was recycled. In the latter case the oil (XRM-177F) was filtered through a series of aircraft and Purolator filters. Oil analysis after these filtering operations showed essentially the same properties as the fresh oil. The recycled oil was used for all of Test Series VI.

A summary of test hours in each test series shows the following:

Test Series	Total No. of Test Hours
I	11,628
II	6,544
III	7,582
IV	1,324
V	11,574
VI	15,450
Total	54,102

Since the entire system was N_2 inerted, there was a question whether it would be necessary to de-aerate the oil prior to its being changed into the bearing tester. Consequently, it was decided that if this became necessary, a relatively large capacity tank would be required which could be used to remove any adsorbed O_2 from the test fluids. The apparatus for accomplishing this is shown schematically in Figure 26 and a photograph of the equipment is shown in Figure 27, Basically, the apparatus consisted of an electrically heated 30-gallon pressure vessel which could be evacuated to a pressure of 10^{-3} torr, The deaeration procedure was as follows:

1. Fill. vessel with 25 gallons of test fluid.
2. Heat to 200F.
3. Reduce pressure to 10^{-3} torr.
4. Hold for 72 hours, agitating slowly during this period.
5. Back-fill tank with high purity N_2.
6. Transfer required quantity to tester oil tank under a nitrogen blanket.

This procedure was used on tests No. 2 through 6 in Test Series I. During this period of time, measurements were made on oil deaerated using the previously described apparatus as well as oil charged directly into the testers without exposure to the deaeration procedure. It was found that within a very short time (approximately 60-90 minutes) the oil placed directly into the testers had been purged of O_2 as efficiently as the oil held for 72 hours at 10^{-3} torr. This assumption was verified by O_2 measurements of the exhaust gas stream as well as the visual appearance of the bearing components and particularly the Monel retainers which would be extremely susceptible to oxidation at the 600F test temperature, The results are not surprising as the oil is: a.) at a very high temperature, b.) vigorously agitated and c.) continually exposed to high purity nitrogen.

In view of the above results, it was decided that it would not be necessary to deaerate the oil prior to placing it into the test units.

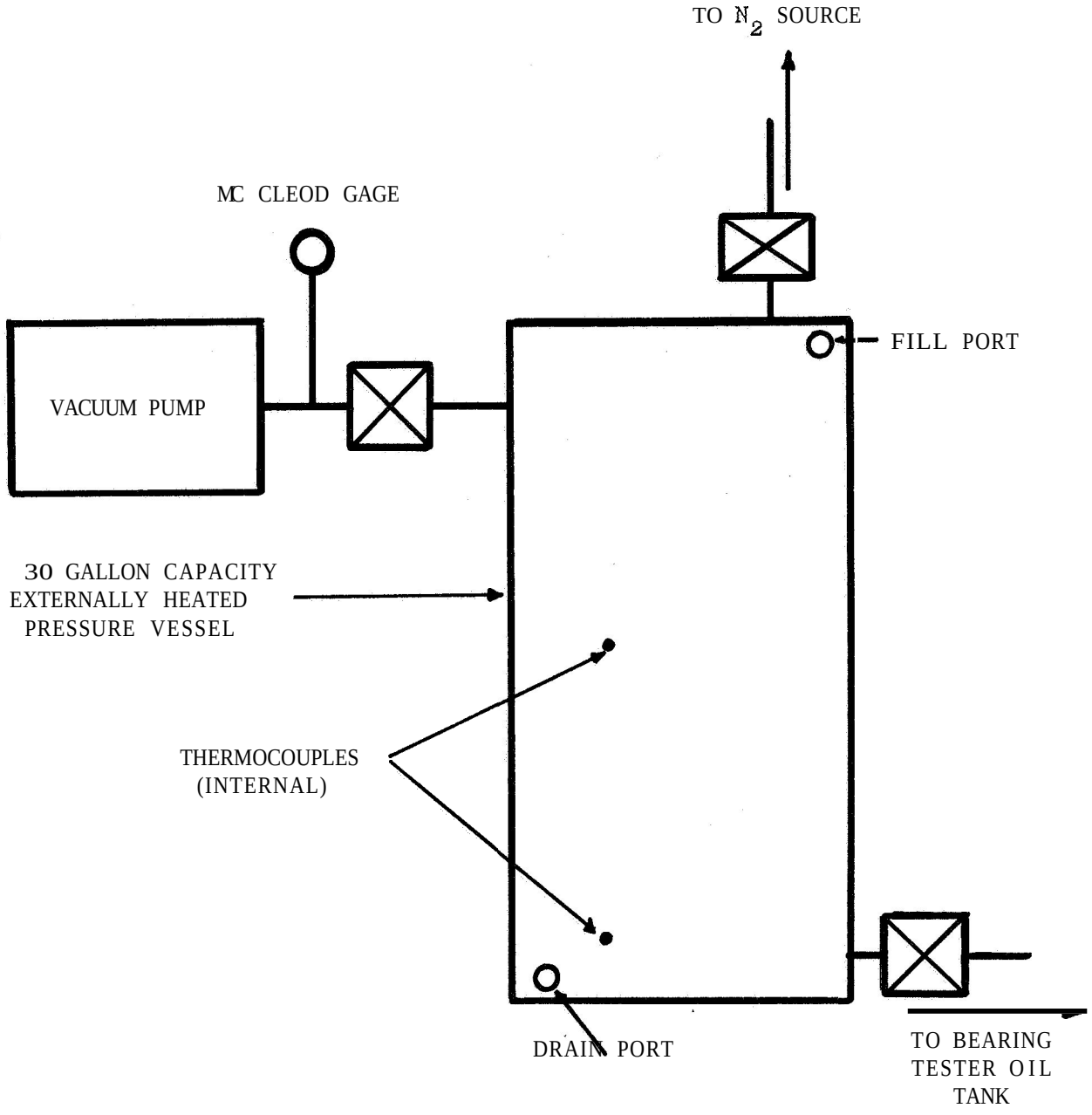
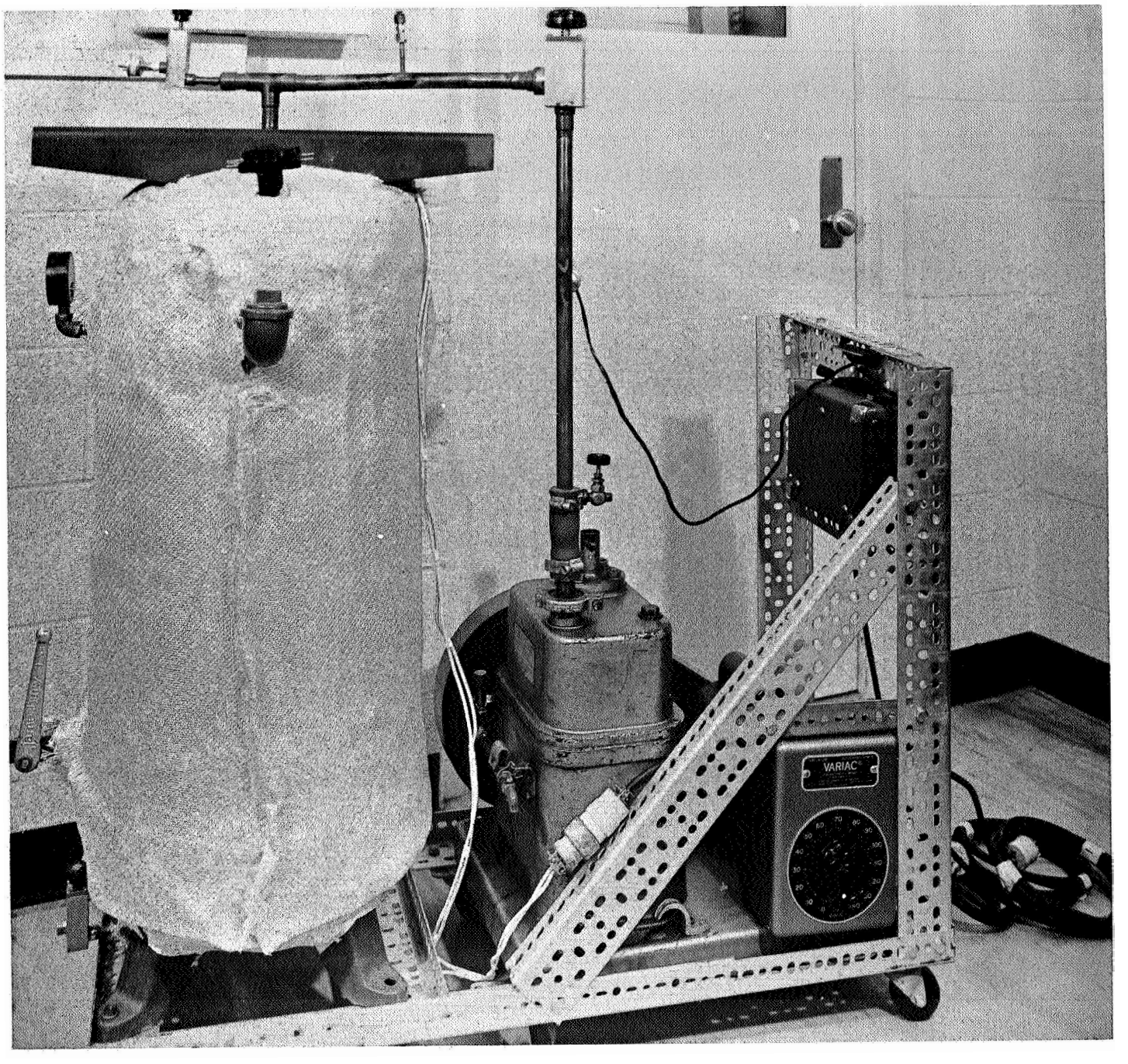


FIGURE 26. SCHEMATIC OF LUBRICANT DEAERATION APPARATUS



FIGURF: 27. APPARATUS FOR DEAERATING **TEST** FLUIDS

6.2.1 Test Series I

The first test series was performed with the materials and test conditions given below:

Bearing Material:	Rings:	CVM-M-50
	Balls:	CVM-M-50
	Retainer:	S-Monel
Lubricant:	Mobil XRM-177	
Operating Conditions:		
	Temp:	Outer Ring 585F ± 5F
		Inner Ring 600F - 610F
		Oil In - 545F-565F
		Oil Out - 585F-600F
	Speed:	12,000 rpm
	Load:	5800 lbs, axial (28 tests)
		7000 lbs. axial (4 tests)
	Atmosphere:	N_2 <5 ppm O_2

The results of these tests are tabulated in Table 12 and the resultant Weibull plot is presented in Figure 28. The Weibull Analysis considers 26 out of the 32 tests, thus carrying a failure index of 6 of 26. Not considered in the analysis are tests #I-2 and I-2a (Bearing #21 and 22) which were damaged during disassembly and tests #I-8, I-8a, I-9, I-9a which were performed at the 7000 lb, load level, The Weibull Analysis is performed using the methods of L. G. Johnson,[8] i.e. the suspensions are included in the calculations. This method was also used for all subsequent analysis.

The failures in Table 12 marked with a rectangle are considered to be normal sub-surface initiated failures. Serial number 20 was an inner ring failure detected just at the point of initiation by an audible noise change. Consequently, the failures were in a very early stage, and mainly because of this, it was decided to allow subsequent failures

Table 12

Results of 120 mm Bearing Tests with XRM-177F Lubricant, CVM M-50 Bearings, at 600F in N_2

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
I - 1	19	5800 (1)	144	OK	OK	OK
I - 1a	20	5800	144	Fatigue	OK	OK
I - 2	21	5800	30	OK	OK	(0005 Wear)
I - 2a	22	5800	30	OK	OK	(0003 Wear)
I - 3	15	5800	309 (3)	OK	Fatigue	Fatigue
I - 3a	16	5800	309 (3)	OK	OK	OK
I - 4	17	5800	290	OK	OK	OK
I - 4a	18	5800	290	OK	OK	Fatigue
I - 5	9	5800	500	OK	OK	OK
I - 5a	10	5800	500	OK	OK	OK
I - 6	11	5800	500	OK	OK	OK
I - 6a	12	5800	500	OK	OK	OK
I - 7	13	5800	500	OK	OK	OK
I - 7a	14	5800	500	OK	OK	OK
I - 8	27	7000 (2)	117	OK	OK	(002 Wear)
I - 8a	28	7000	117 (Cage Failure)	OK	OK	(003 Wear)
I - 9	25	7000	165	OK	OK	(0005 Wear)
I - 9a	26	7000	165	OK	OK	(002 Wear)
I -10	17	5800	375	OK	OK	Fatigue
I -10a	16	5800	434	OK	OK	OK
I -11	33	5800	189	Fatigue	OK	OK
I -11a	34	5800	189	OK	OK	OK

Table 12, Contd.

Test No.	Bearing S/N	Load (lb.),	Time (Hr.)	Inner Ring	Outer Ring	Ball
I -12	23	5800	519	OK	OK	OK
I -12a	24	5800	519	OK	OK	OK
I -13	29	5800	500	OK	OK	OK
I -13a	30	5800	500	OK	OK	OK
I -14	31	5800	525	OK	OK	OK
I -14a	32	5800	525	OK	OK	OK
I -15	37 (4)	5800	396	OK	OK	OK
I -15a	38 (4)	5800	396	OK	OK	OK
I -16	35	5800	518	OK	OK	OK
I -16a	36	5800	518	OK	OK	OK
I -17	42	5800	465	OK	OK	OK
I -17a	41	5800	465	Fatigue	OK	OK
I -18	16	5800	630	OK	OK	OK
I -18a	34	5800	385	Track Damage	Track Damage	OK

(1) 5800 lb. axial load = 321,000 psi max. hertz on inner.

(2) 7000 lb. axial load = 342,000 psi max. hertz on inner.

(3) Time to initiation of outer ring spall. Bearing run an additional 40 hours after spall initiation.

(4) Severe misalignment.

☐ Indicates Time to Fatigue Failure

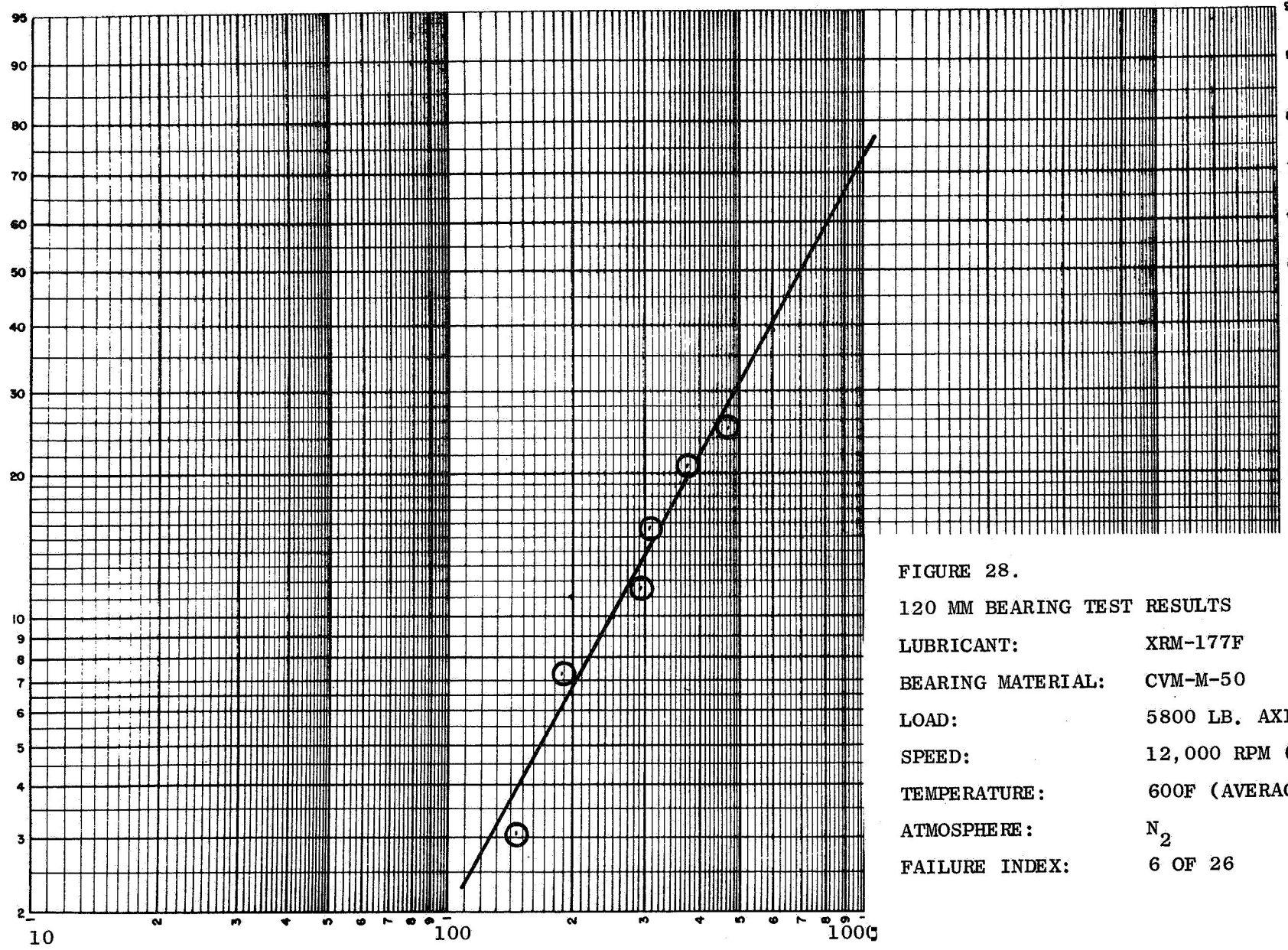


FIGURE 28.

120 MM BEARING TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	CVM-M-50
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	600F (AVERAGE METAL)
ATMOSPHERE:	N_2
FAILURE INDEX:	6 OF 26

to progress to a somewhat larger spall before terminating the test. The running mate of this bearing, serial number 19, showed evidence of some inner ring damage, although it did not appear to be due to fatigue, It is thought that the damage observed may be due to foreign objects in the oil.

Bearing 15 was believed to have had its failure origin in an outer ring spall. Since this was the first positively identifiable fatigue failure, the bearing was run for an additional 40 hours over the 309 hours required to initiate the outer ring spall in order to investigate vibration level change as a function of time. As a result of this, ball failures, evidenced by severe spalling, were encountered and finally a cage web break caused an uncontrollable temperature rise which forced the shut-down of the test. An overall view of this bearing is shown in Figure 29 and a close-up of the cage failure in Figure 30. The initiating spall in the outer ring is shown in Figure 31, whereas Figure 32 shows the severely damaged balls. The assumption of an outer ring spall being the initiating failure is based on the texture of the spalls. It will be noted that the surface of the outer race failure is relatively smooth compared to the rather jagged appearance of most of the failed areas on the balls. The inference here is that the earlier failure, due to continual ball passage, has a surface smoother than the newer failures. In this particular failure, the vibration level did not increase sufficiently to cause automatic shutdown of the equipment, although had the over temperature controls not been bypassed, the machine would have been shut-down shortly after the appearance of the initial spall on the outer ring.

Bearing 18 exhibited a ball failure after 290 hours of operation. This ball failure is shown in Figure 33 along with several other balls from the bearing showing the generally good appearance of these components. Figure 34 shows the inner ring of this bearing, the arrow indicating the dimensions of the ball path. Figure 35 is a view of the cage pockets of this bearing showing the minimal amount of cage wear which has occurred in this period.

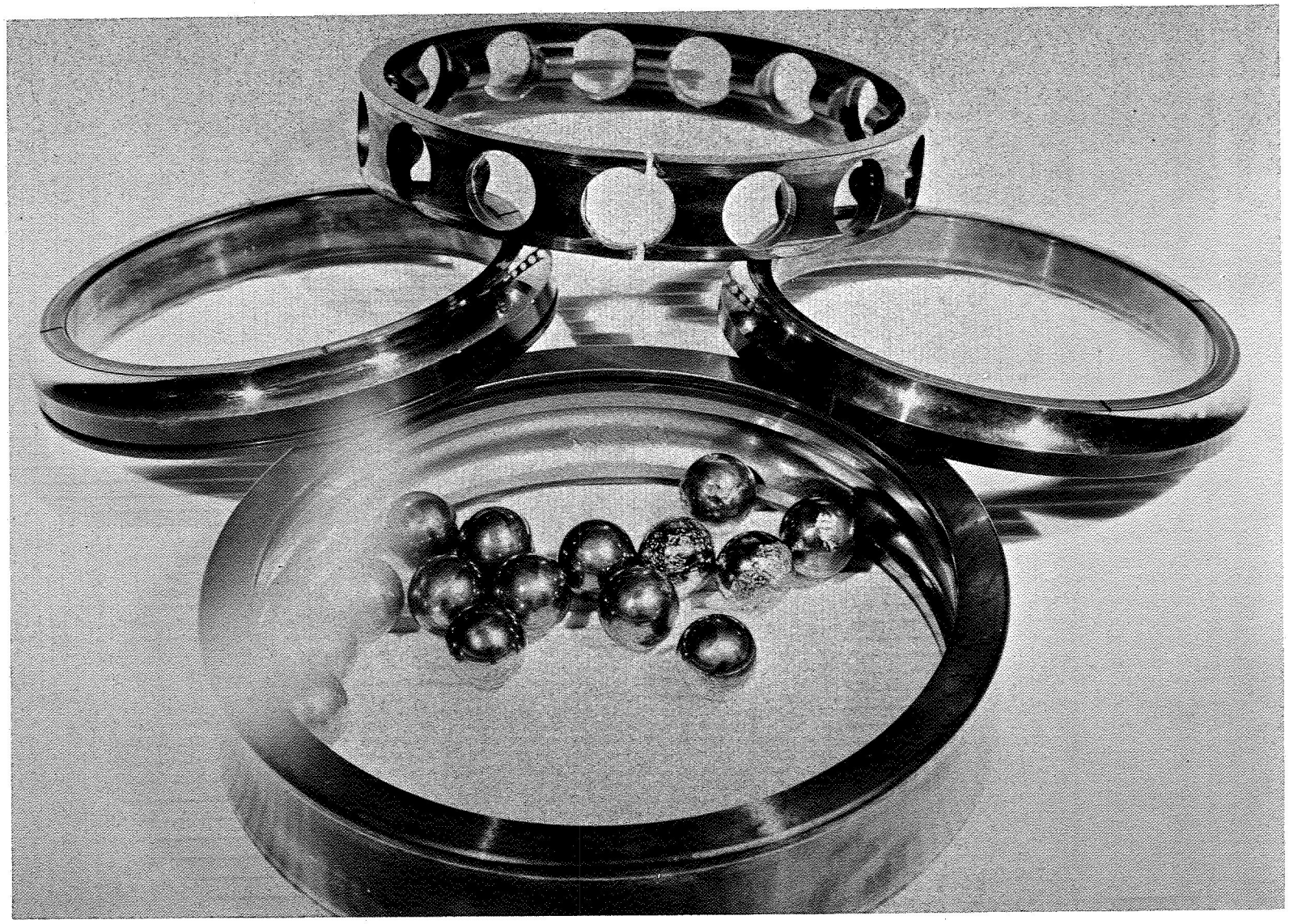


FIGURE 29. OVERALL VIEW OF BEARING S/N 15 AFTER 349 HOURS

RUNNING TIME:	349 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

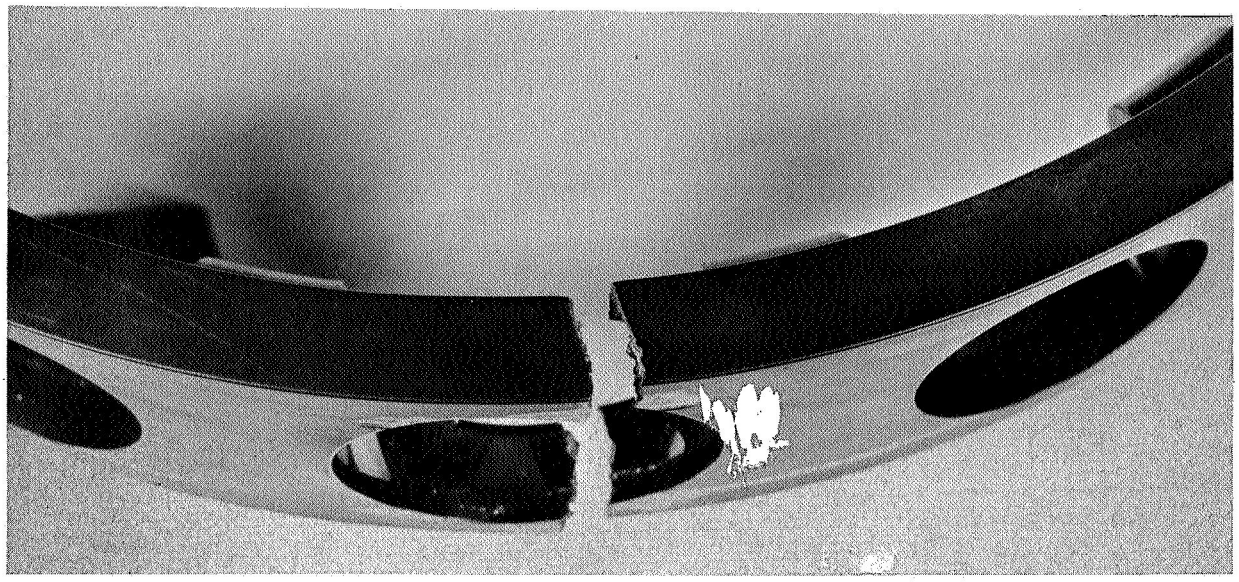


FIGURE 30. CAGE FAILURE - BEARING S/N 15

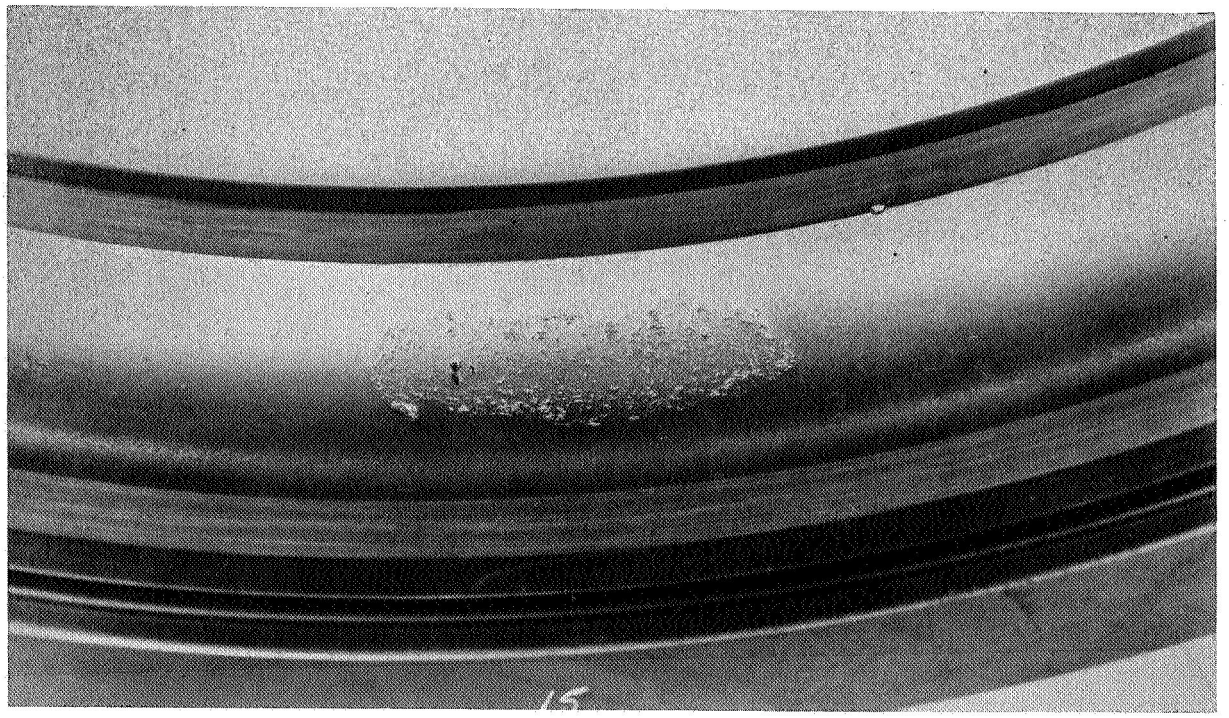


FIGURE 31. FATIGUE FAILURE - OUTER RING - BEARING S/N 15

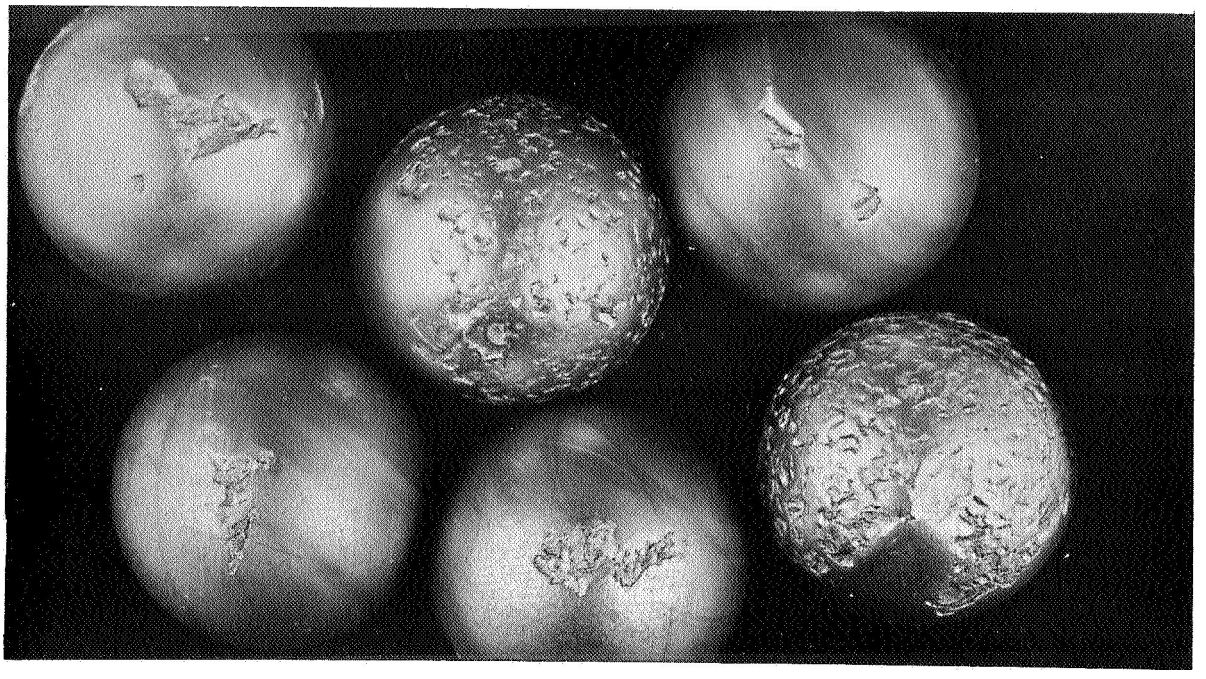


FIGURE 32. BALL FAILURES - BEARING S/N 15

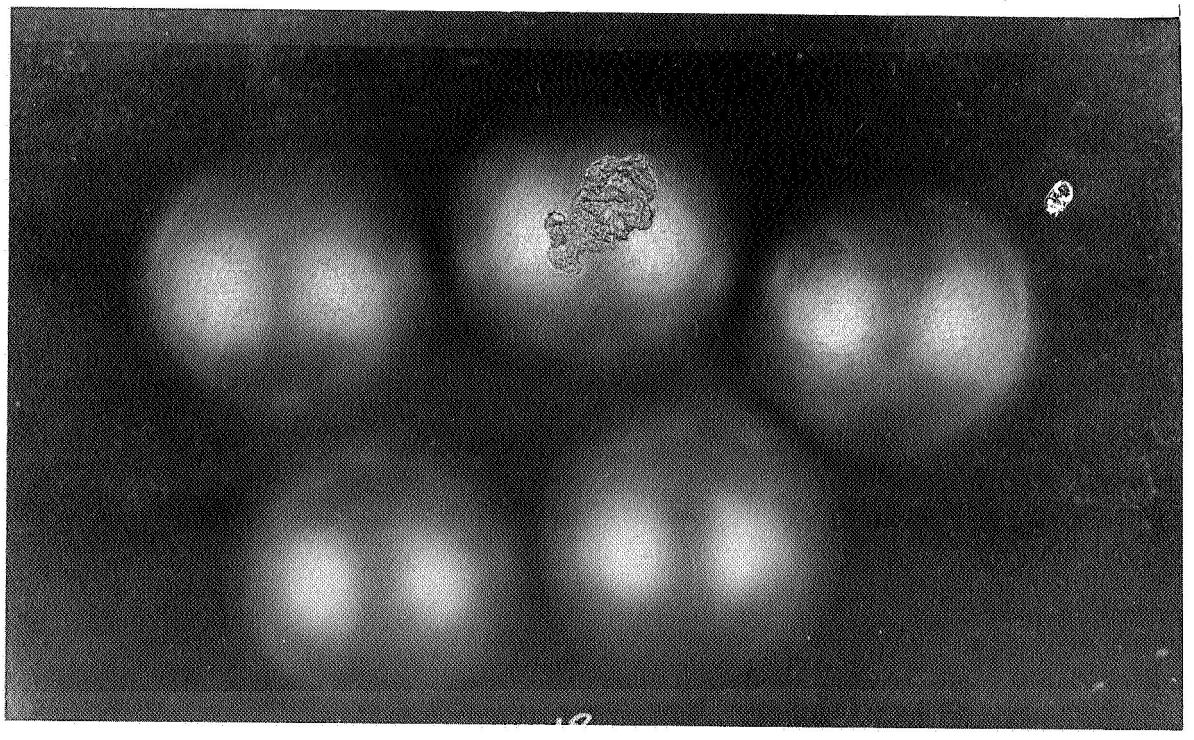


FIGURE 33. BALL FATIGUE FAILURE - BEARING S/N 18

RUNNING TIME:	290 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

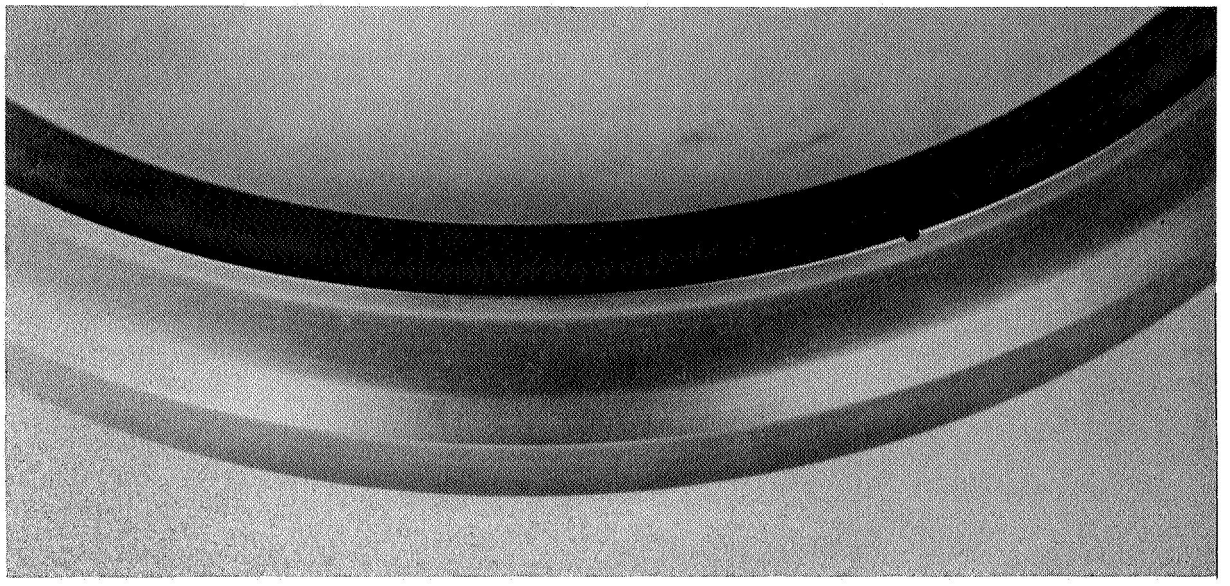
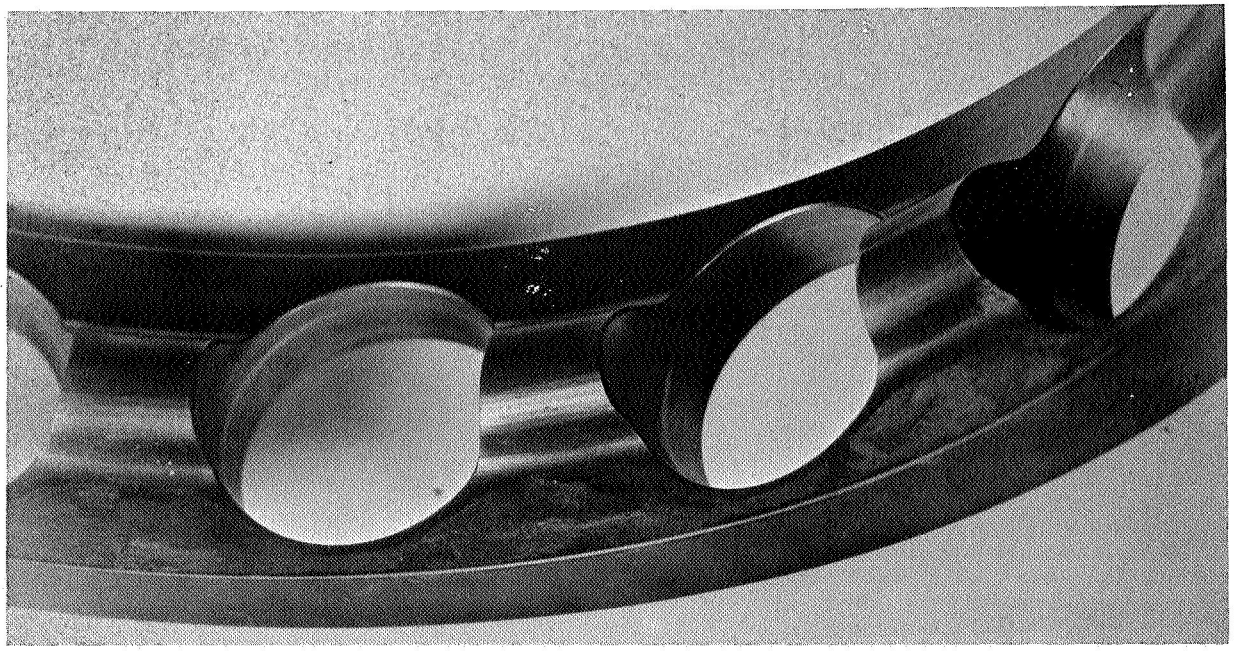


FIGURE 34. INNER RING BALL PATH - BEARING S/N 18



FIGURF: 35. CAGE BALL POCKETS - BEARING S/N 38

An interesting observation can be made regarding the occurrence of the ball failures. In normal bearing fatigue testing, the probability of ball failures is relatively low. In the present case, however, it had been noted in the RC Rig tests that the rolling contact fatigue life of the M-50 CVM billet used for the balls was lower than the ring material. In view of this, a greater incidence of ball failures had been expected, and this expectation appears to have been correct.

Bearing 17 failed after 375 hours. The failure was characterized by a fatigue failure on the outer ring, as well as three ball failures. The failed balls are shown in Figure 36, and the failed outer ring in Figure 37. Bearing 33 had a massive inner ring fatigue failure after 189 hours. The failed ring is shown in Figure 38.

A number of bearings achieved 500 hours. One of these was Bearing #10 and Figure 39 shows several balls from this bearing,. The outer ring ball track of the same bearing is shown in Figure 40.

An overall view of another 500 hour bearing, serial number 13, is shown in Figure 41 with the next several photographs showing the appearance of the balls (Figure 42), retainer (Figure 43) and inner and outer ball paths (Figures 44 and 45). Considering the extremely severe conditions to which these bearings have been exposed, it is highly encouraging to note the almost undamaged appearance of these various components.

At the 5800 lb. load level, the semi-major axis of the pressure ellipse on the inner race is calculated to be .085 inches. The measured wear track is .100 inches, somewhat greater than calculated. This is, however, not unexpected based on the rolling contact fatigue testing and other bearing testing experience.

The only cases of ball wear at the 5800 lb. load level with the XRM-177F occurred with bearings 21 and 22. The balls in both of these

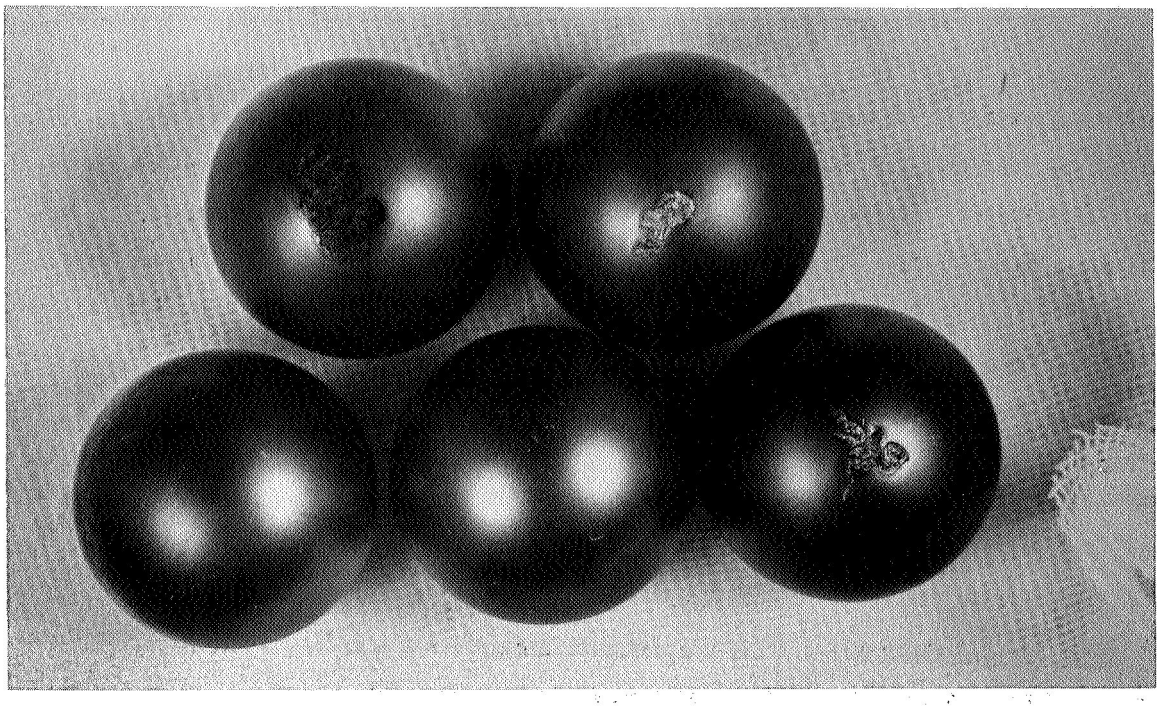


FIGURE **36.** BALL FAILURES - BEARING S/N 17

RUNNING TIME:	375 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

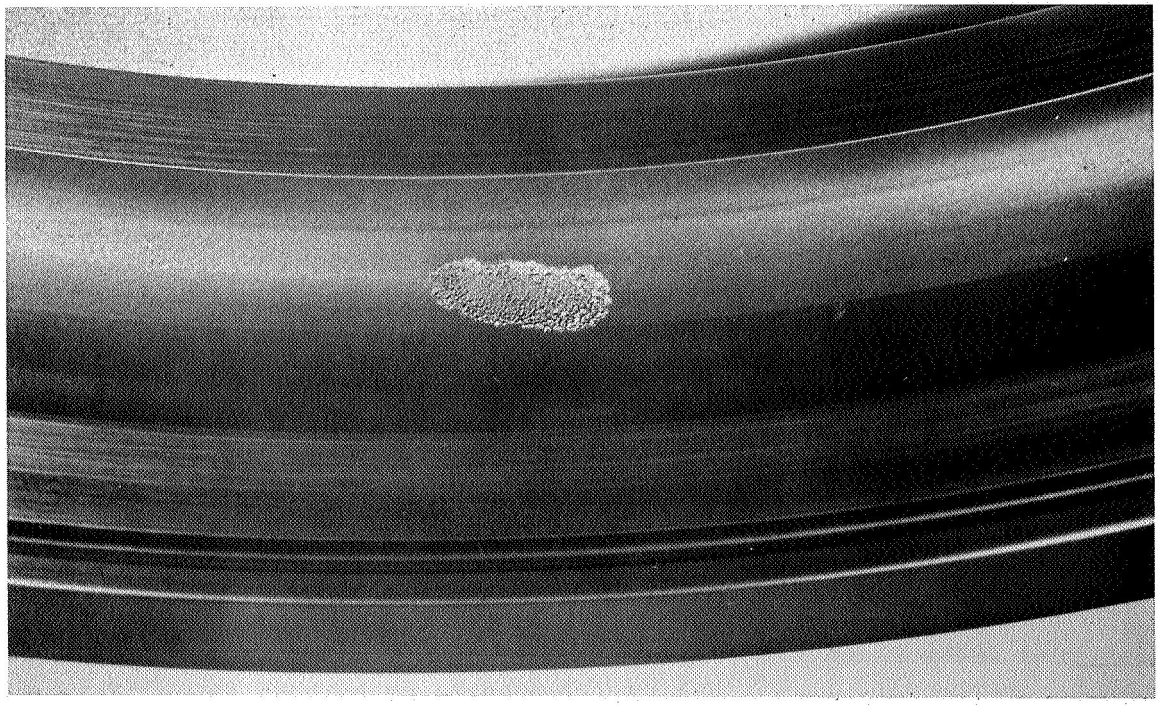


FIGURE 37. FATIGUE SPALL ON OUTER RING OF **BEARING S/N** 17

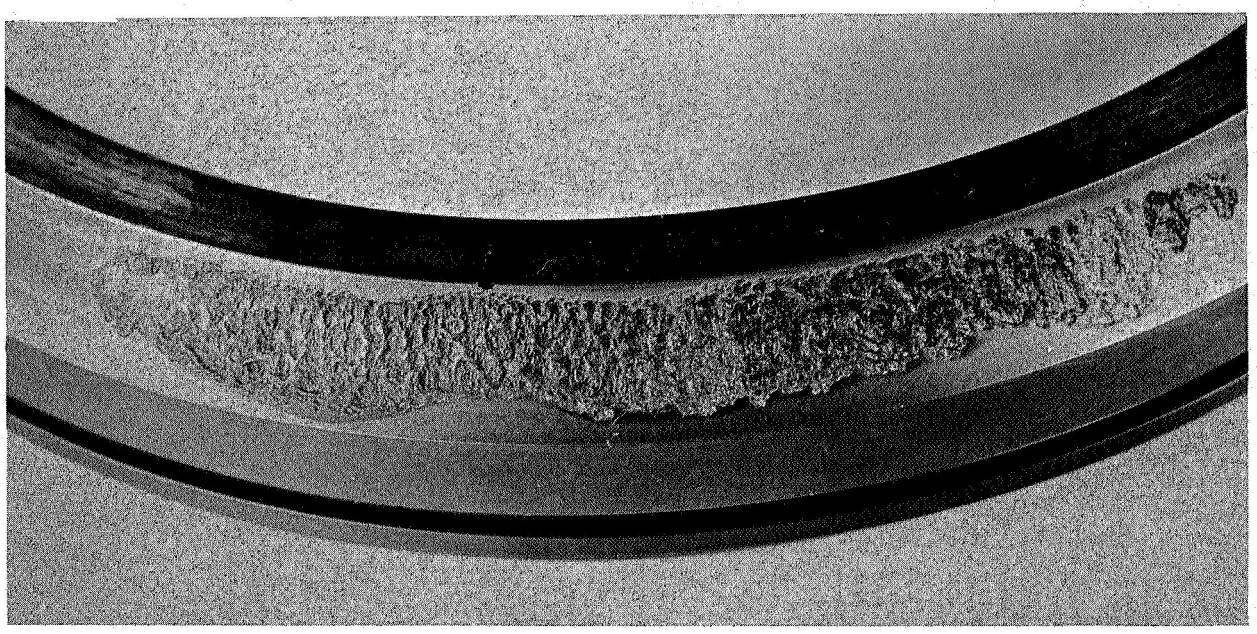


FIGURE 38. FATIGUE SPALL ON INNER RING - BEARING S/N 33

RUNNING TIME:	189 HOUR$
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

FIGURE 39. BALLS FROM BEARING S/N 10

RUNNING TIME:	500 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

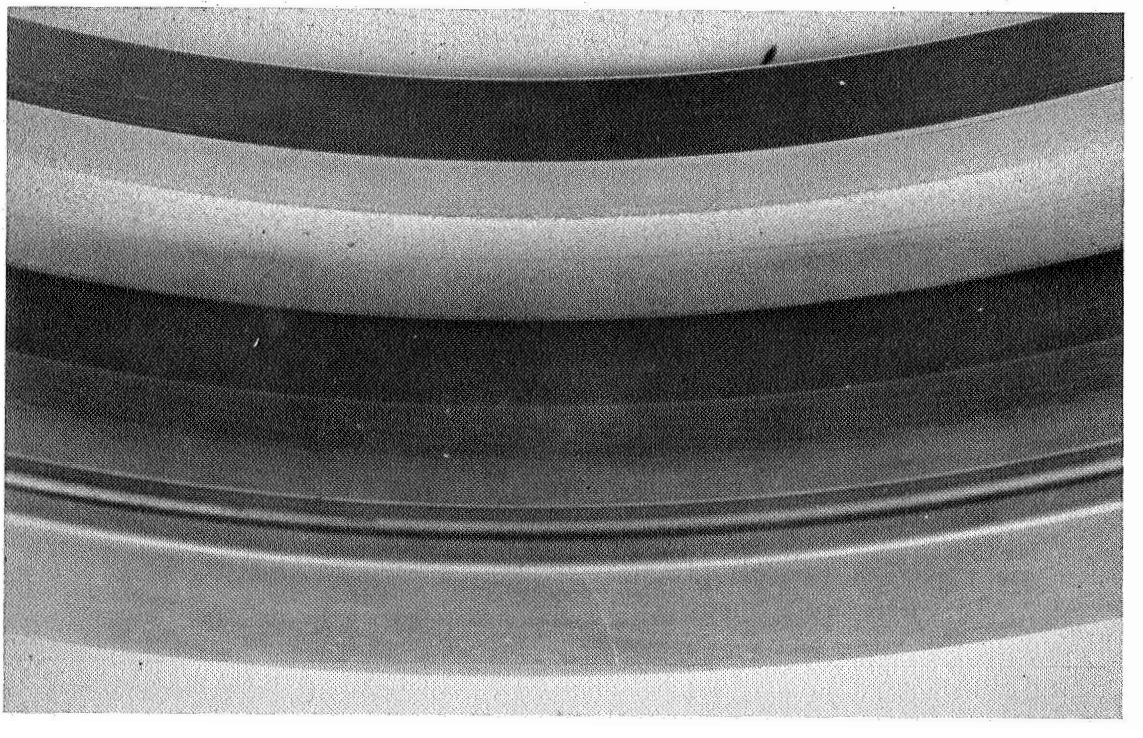
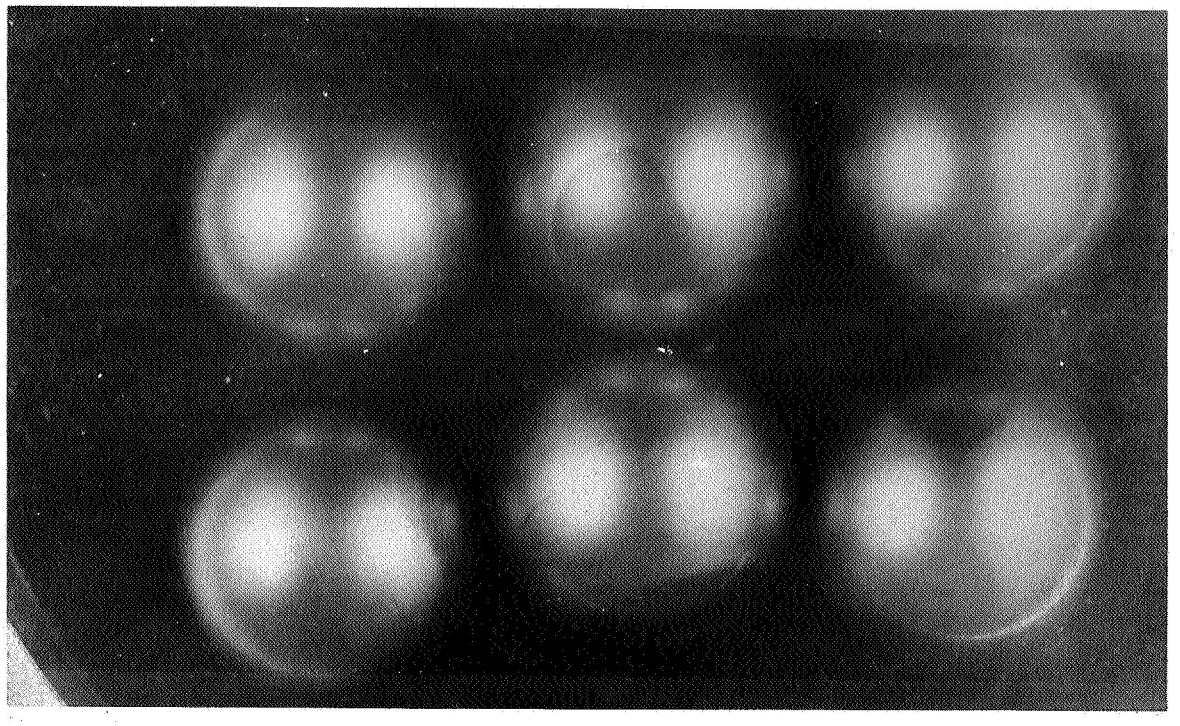


FIGURE 40. OUTER RING - BALL PATH BEARING S/N 10

FIGURE 41. OVERALL VIEW OF BEARING S/N 13 AFTER 500 HOURS

RUNNING TIME:	500 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)



FIGURF: 42. BALLS FROM BEARING S/N 13

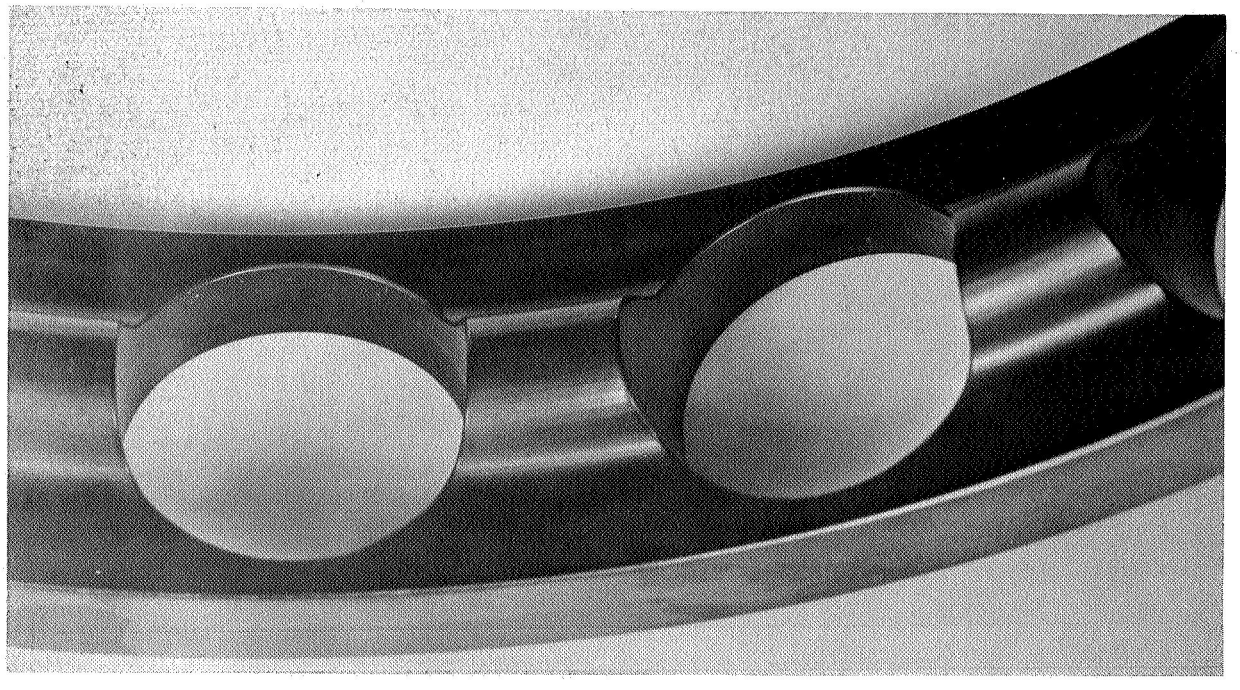


FIGURE 43. CAGE BALL POCKETS - BEARING S/N 13

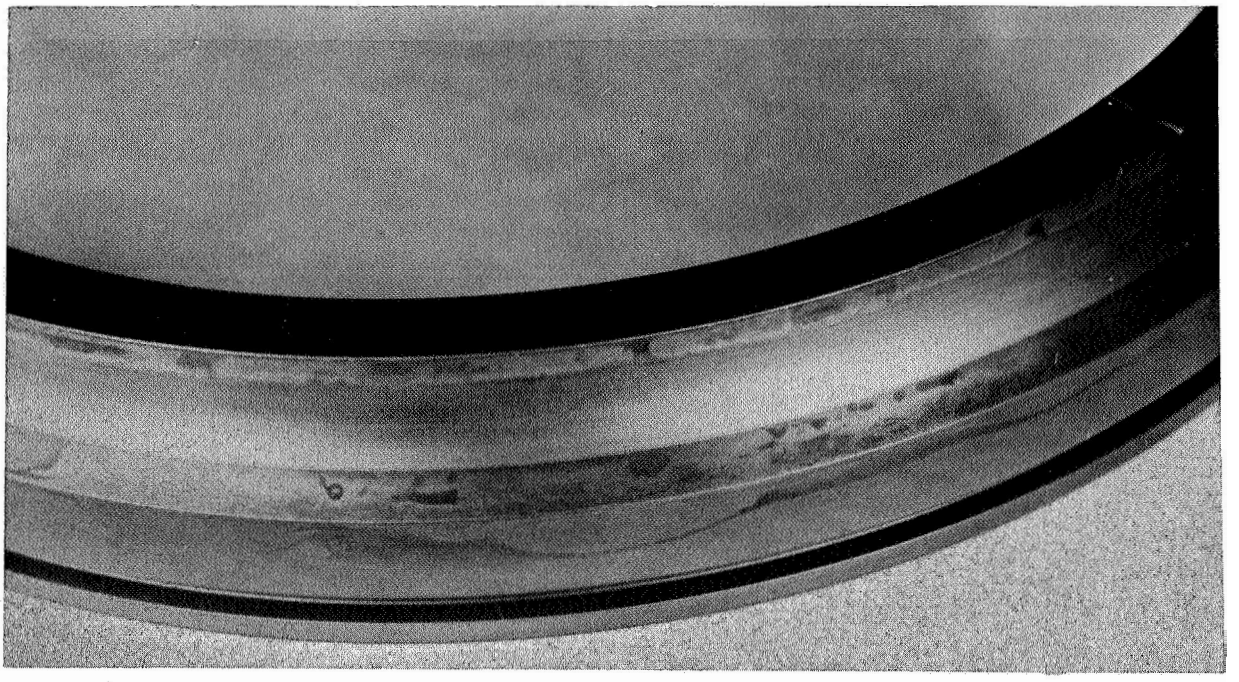


FIGURE 44. INNER RING BALL PATH - BEARING S/N 13

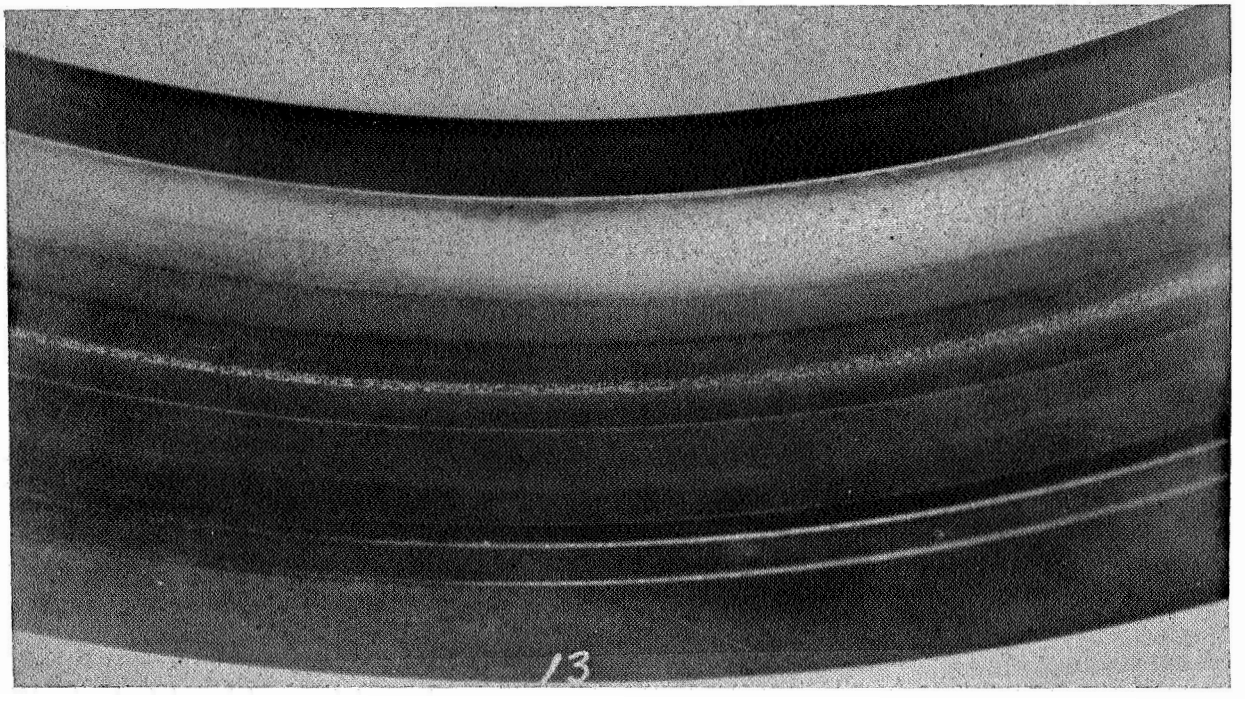
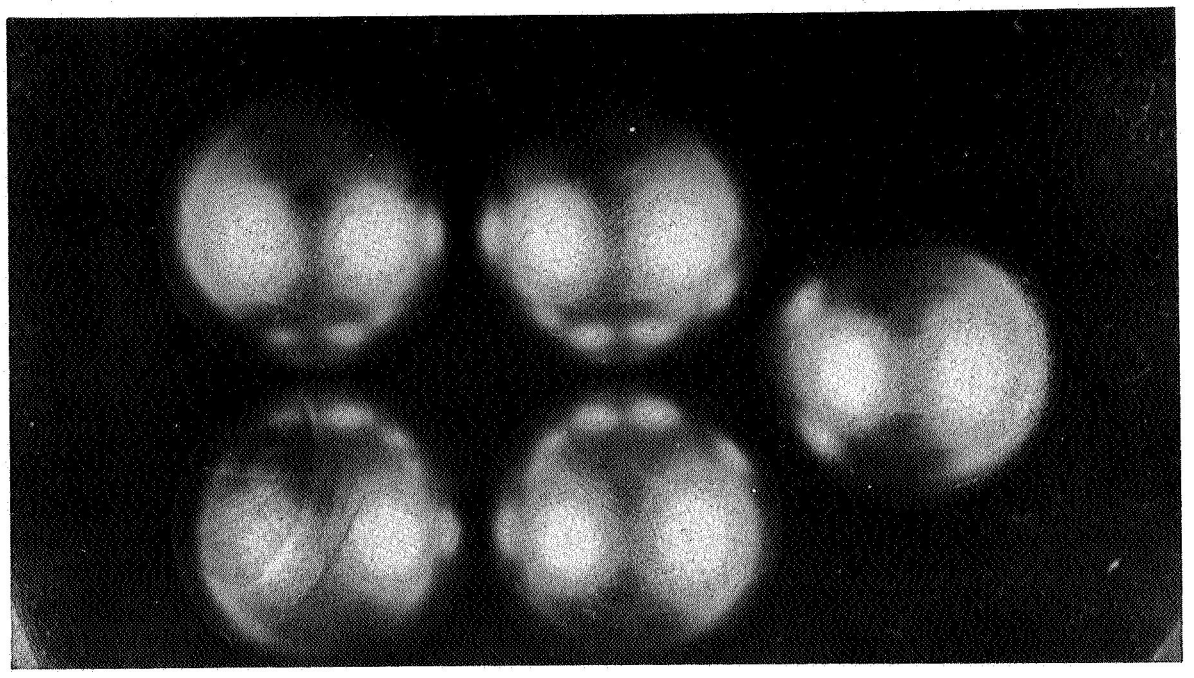


FIGURE 45, OUTER RING BALL PATH - BEARING S/N 13

bearings, unlike all other runs, showed micro pitting similar to that observed on the bearings run subsequently with the polyphenylether fluid, The ball pitting is shown in the photograph in Figure 46 for bearing 21, The remaining bearing components were essentially undamaged. For both of these bearings, it was determined that no misalignment had existed, and the lubricant was in good condition after test, No explanation could be found for the short (30.3 hours) running time and the rapid' ball wear which ranged from .0003" - .0005".

Four bearings, serial numbers 25 through 28 inclusive, were run at an axial load of 7000 lb. in order to investigate the effects of higher loading. Bearings 27 and 28 were inspected after 100 hours of operation and found to be in good condition. No component wear was observed at this time, These two bearings were reassembled and ran for an additional 17 hours. At this point, the bearings were shut-down due to over-temperature indications, Subsequent examination of the bearings showed a cage failure and a high degree of ball wear on the order of .0005". The cage was severely deformed, and cracks were observed in all of the side rails on one side. Bearings 25 and 26 ran for 165 hours before they were shut-down by over-temperature indications Measurements showed ball wear to be on the same order as in the first test. The cage, however, appeared to be in excellent condition with no evidence of plastic deformation or cracking.

The next five photographs, Figures 47 through 51, show views of bearing #28 run at 7000 lb. axial load, which is equivalent to a maximum Hertzian stress of 342,000 psi on the inner ring. The primary failure in this bearing was the severe cage rupture shown in Figure 47, with a number of secondary failures in the cage web as illustrated in Figure 48 In addition to this, severe wear was encountered on the balls. Typical balls from this bearing are shown in Figure 49. Figures 50 and 51 illustrate the condition of the inner and outer raceways.



'FIGURE 46. WEAR AND MICRO PITTING ON BALLS - BEARING S/N 21

RUNNING TIME:	**30.3** HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

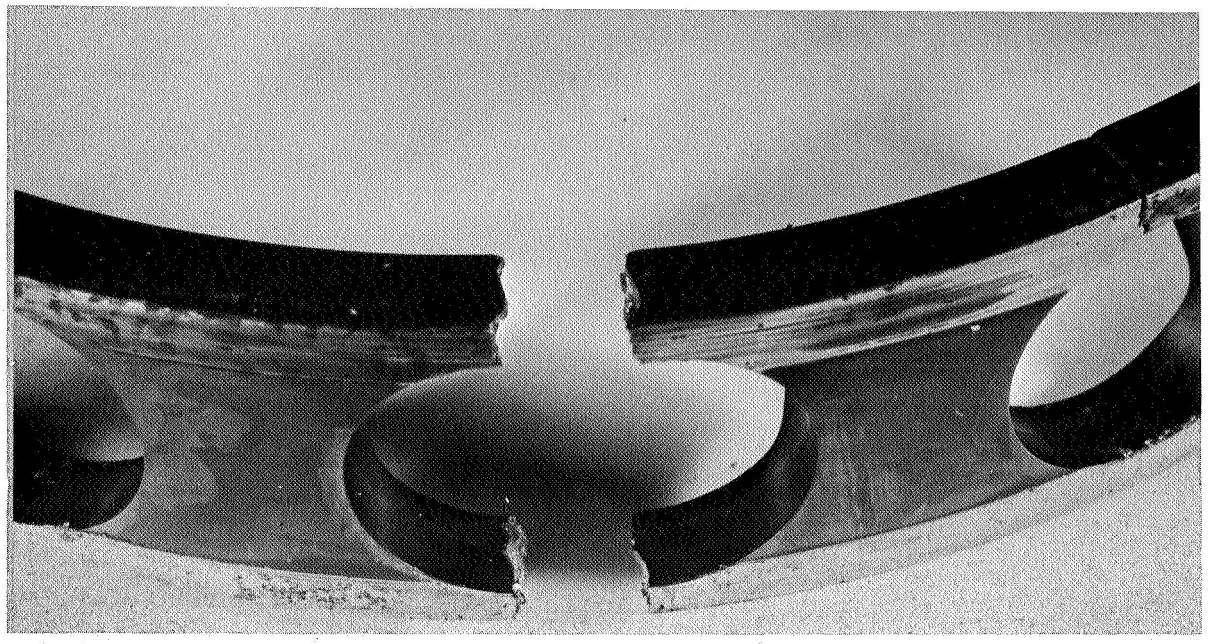
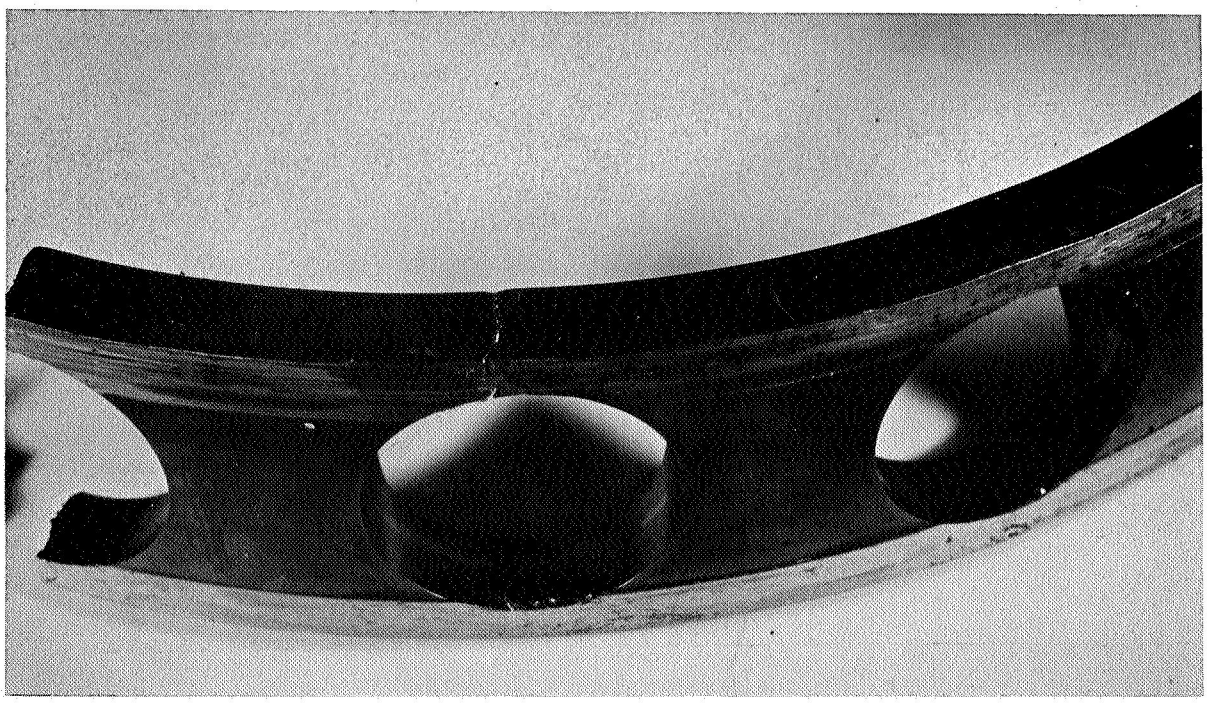
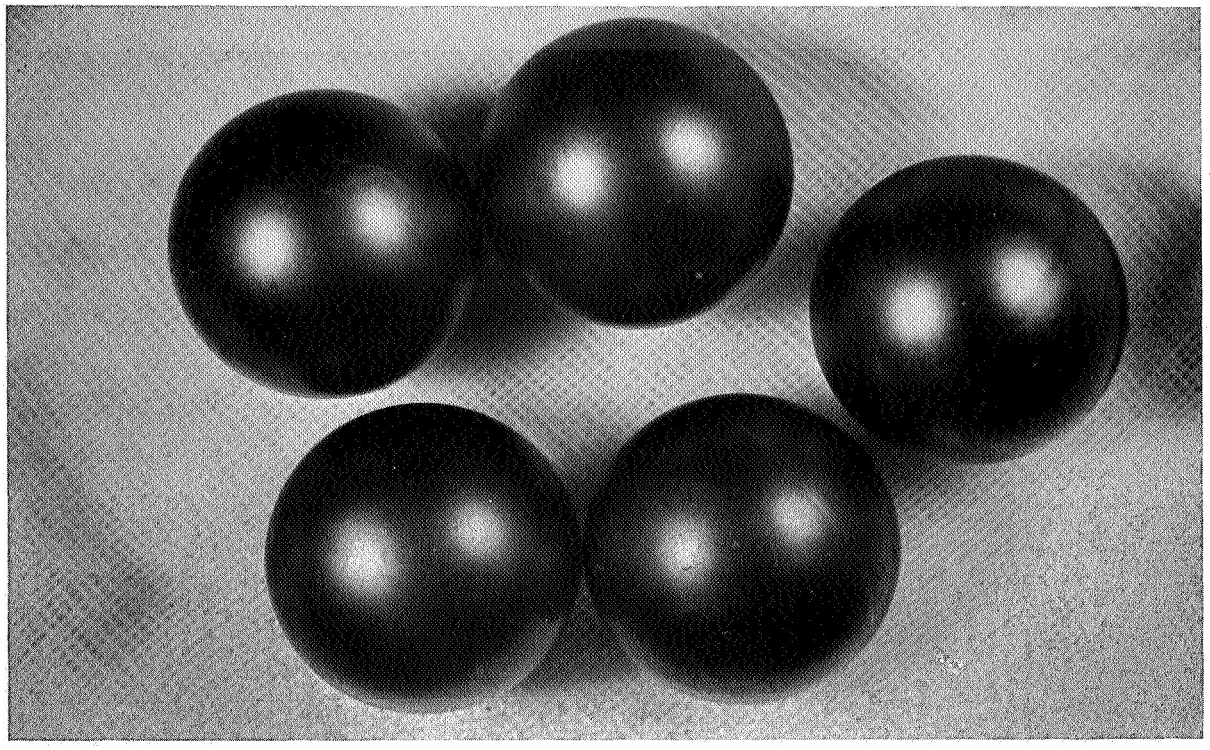


FIGURE 47. PRIMARY CAGE FAILURE - BEARING S/N 28

RUNNING TIME:	117.5 HOURS
LUBRICANT:	XRM-177F
TEMPERATURE:	600F
LOAD:	7000 LB. AXIAL
SPEED:	12,000 RPM (INNER)



FIGURF: 48. SECONDARY RETAINER FAILURE - BEARING S/N 28



FIGURF: 49. TYPICAL APPEARANCE OF BALLS FROM BEARING S/N 28

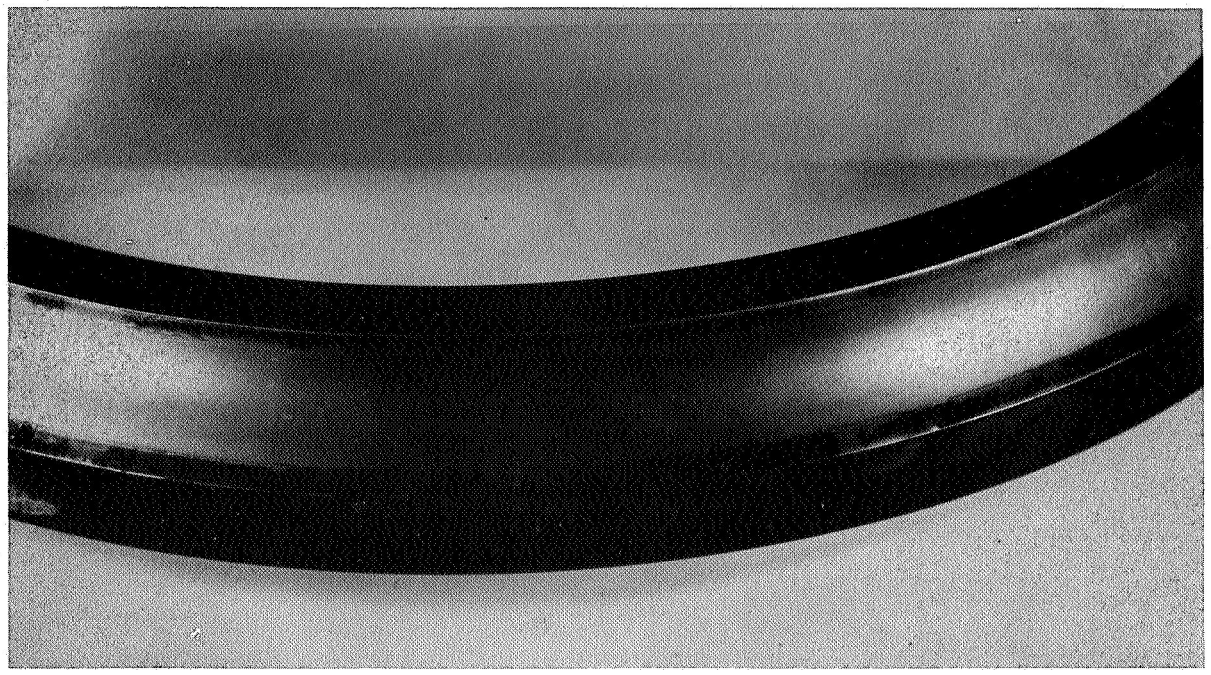


FIGURE 50. INNER RING - BEARING S/N 28

FIGUFE 51. OUTER RING - BEARING S/N 28

NOTE: DAMAGE TO SHOULDER **ABOVE** BALL TRACK.

The implication of these results on the conduct of the test program has previously been reviewed.

The next three photographs, Figures 52, 53, and 54 are views of the inner and outer rings and balls respectively, from bearing 24 after 519 hours of running.

Lastly, Figures 55 and 56 illustrate the "Track Damage" observed on the inner and outer rings of bearing 34. The damage, as such, appears to be due to contamination of the lubricant, although an analysis of the fluid failed to reveal any significant evidence of foreign material in the oil.

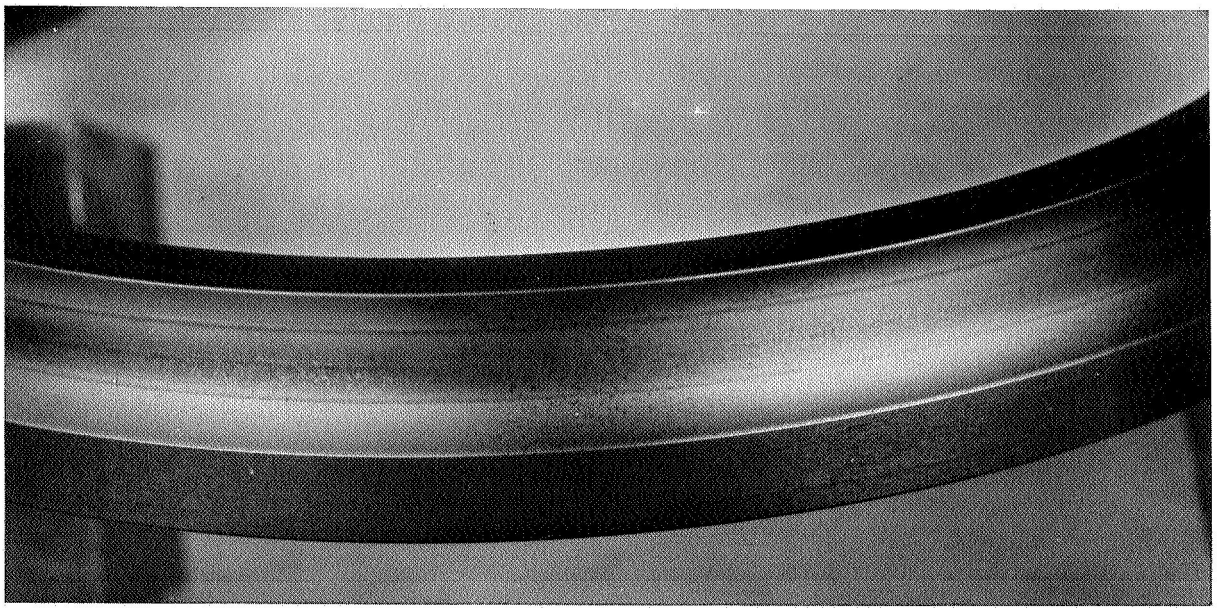


FIGURE 52. BALL PATH OF INNER RING - BEARING S/N 24

RUNNING TIME:	519 HOURS
LUBRICANT :	XRM-177F
TEMPERATURE :	600F
LOAD :	5800 LB. AXIAL
SPEED :	12,000 RPM (INNER)

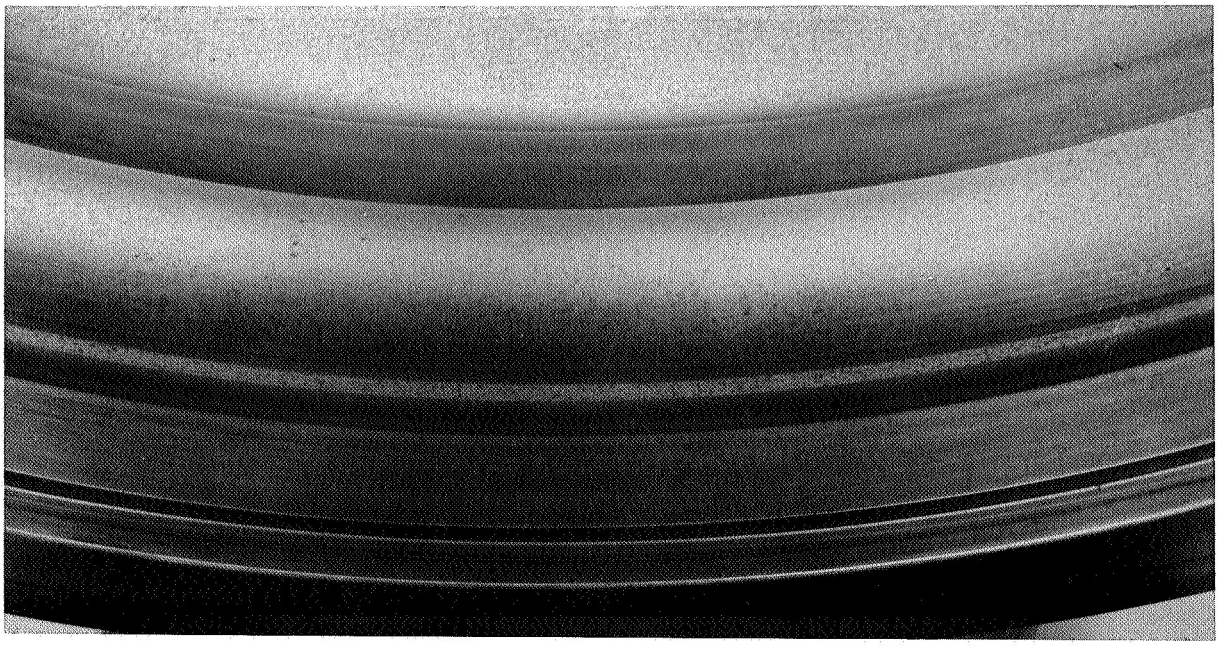


FIGURE 53. BALL PATH ON OUTER RING - BEARING S/N 24

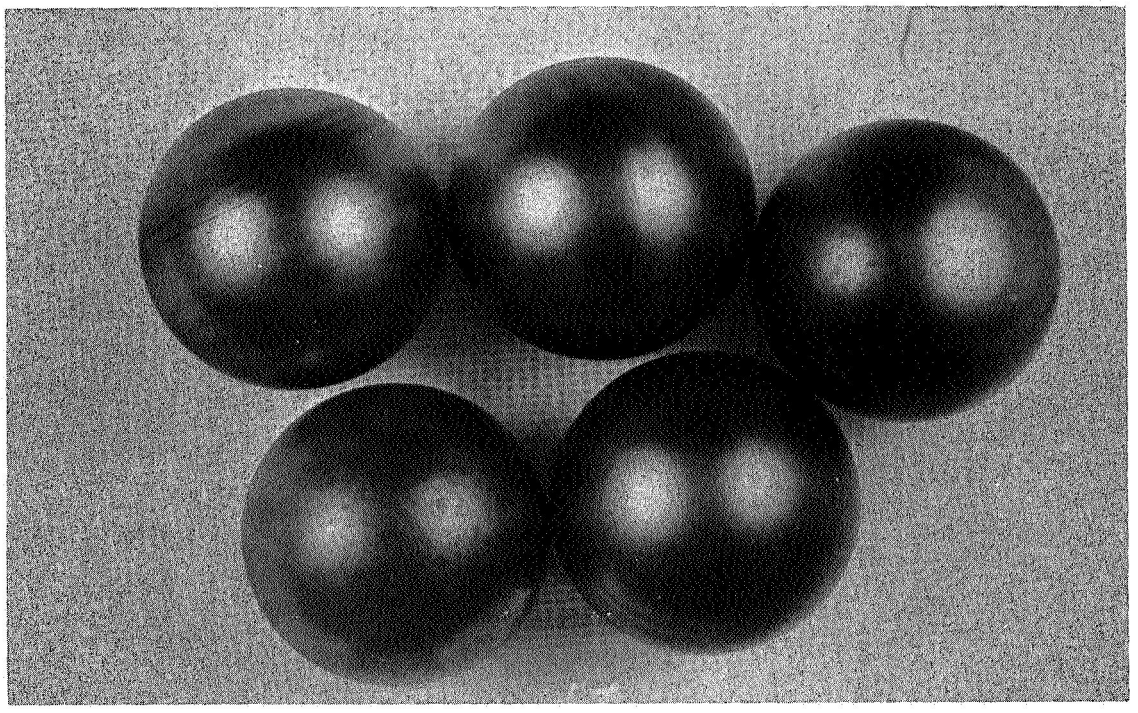


FIGURE 54. BALLS FROM BEARING S/N 24

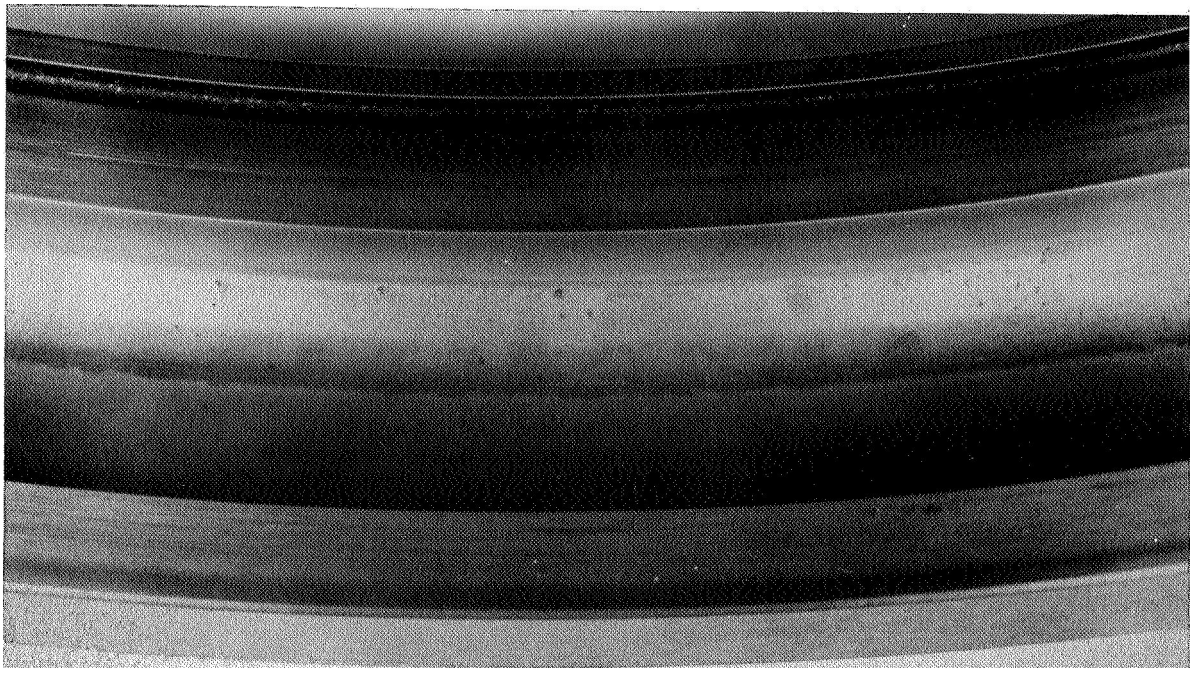


FIGURE 55. OUTER RING TRACK DAMAGE - BEARING S/N 34

RUNNING TIME:	385 HOURS
LUBRICANT:	XRM-177F (N_2 ATMOSPHERE)
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

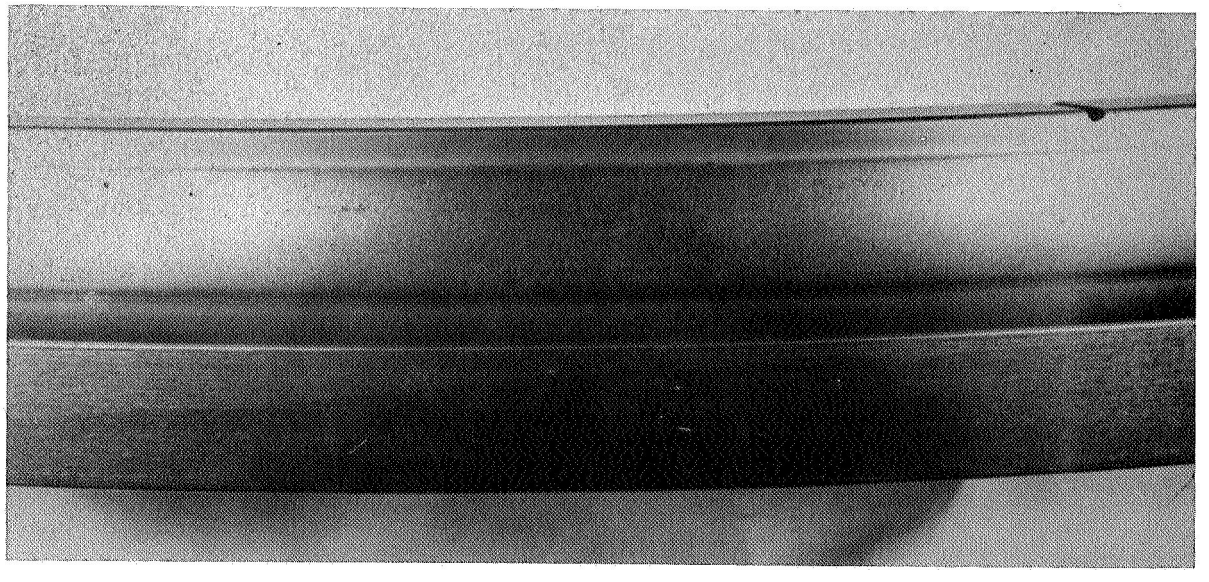


FIGURE 56. INNER RING TRACK DAMAGE - BEARING S/N 34

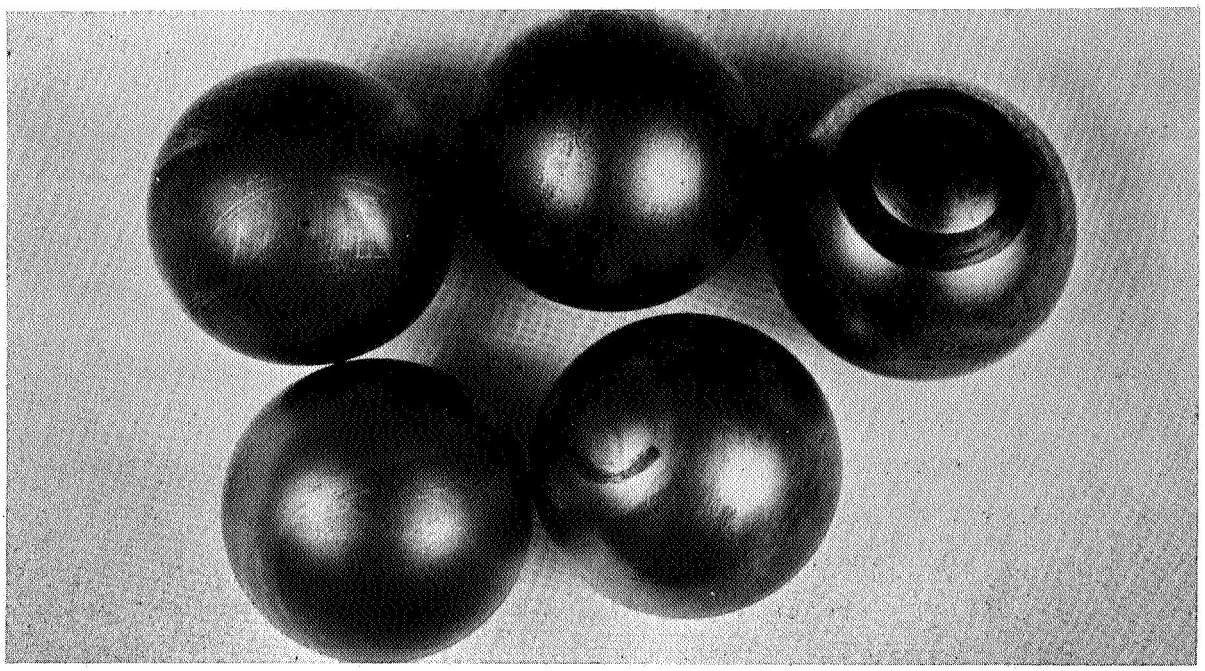
6.2.2 Test Series II

The initial plan for Test Series II was to perform 30 tests, using CVM-M-SO bearings, MCS-354 fluid at 600F in a N_2 atmosphere. The applied load was to be the same as for the mineral oil tests or 5800 lb. axial which is equivalent to a maximum Hertzian load of 325,000 psi.

Earlier RC Rig testing indicated that the MCS-354 would have a lower fatigue life than the XRM-177F. In addition, some evidence of surface distress was noted on the RC Rig bars, The first attempt to operate the bearings with the MCS-354 in a N_2 atmosphere at a 5800 lb. axial load resulted in a shut-down after two hours due to the inability to stabilize the temperature accompanied by a very high noise level, The high noise level, incidentally, was always a signal of wear and/or fatigue failures. Disassembly of the bearing showed considerable ball wear, track damage and catastrophic cage failure in bearing S/N 54, These items are illustrated in Figures 57 through 60. In view of this, a halt was called to the testing and the situation reviewed with NASA personnel, It was decided to reduce the load on the bearings, and using the RC Rig data, a number of RECAP analyses were made, the results of which indicated the bearings should be able to attain a B_{10} life of 100 hours at an axial load of 4365 lbs. or 295,000 psi S_{max}.

A test was initiated and during the 45 minutes of operation, it was not possible to maintain the temperature at 600F. The outer ring temperature continued to climb above 600F. After 45 minutes of operation, the tester was again shut down due to activation of the over-temperature sensor,

Disassembly of the bearing showed fatigue failures around the entire circumference of the inner raceway of bearing 39 and the inner and outer raceway of bearipg 40. Examples of these failureq aye shown in Figures 61 and 62. In additioa, the balls Prom both of these bearings exhibited severe wear; typical balls from bearing 40 are shown in Figure 63. The cages from both bearings were relatively undamaged as shown in Figure 64.



FIGURF: 57. TYPICAL APPEARANCE OF BALLS - BEARING S/N 54

RUNNING TIME:	2 HOURS
LUBRICANT:	MCS-354 (AIR ATMOSPHERE)
TEMPERATURE:	600F +
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)

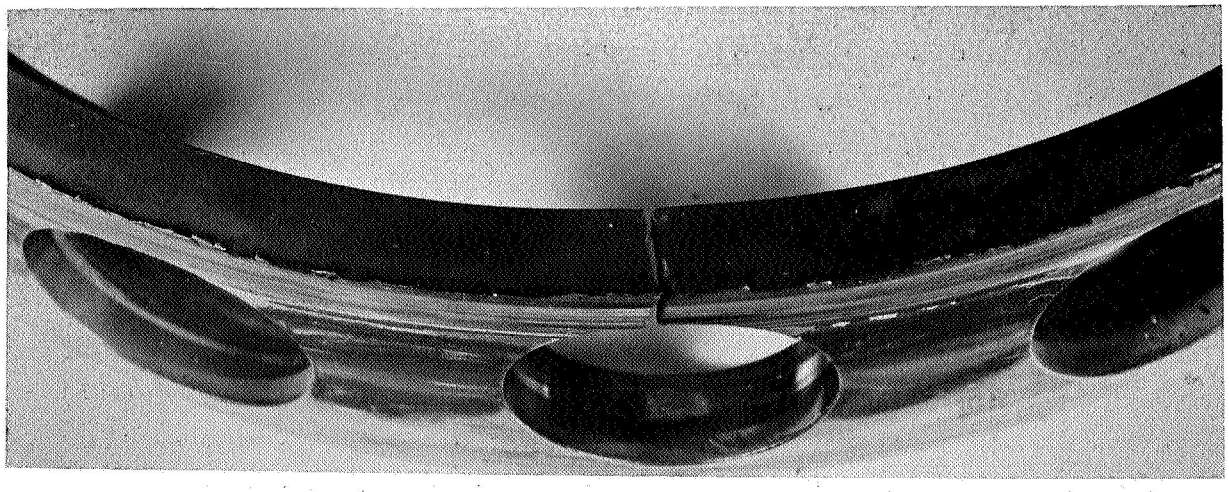
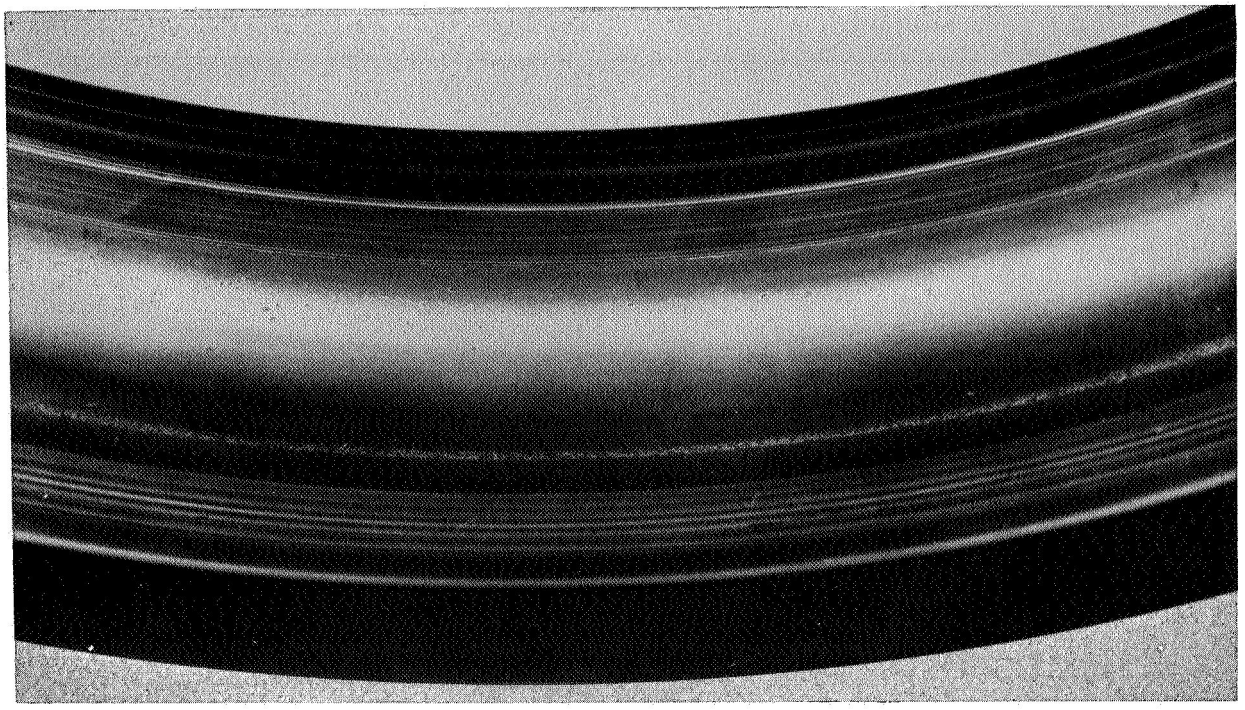
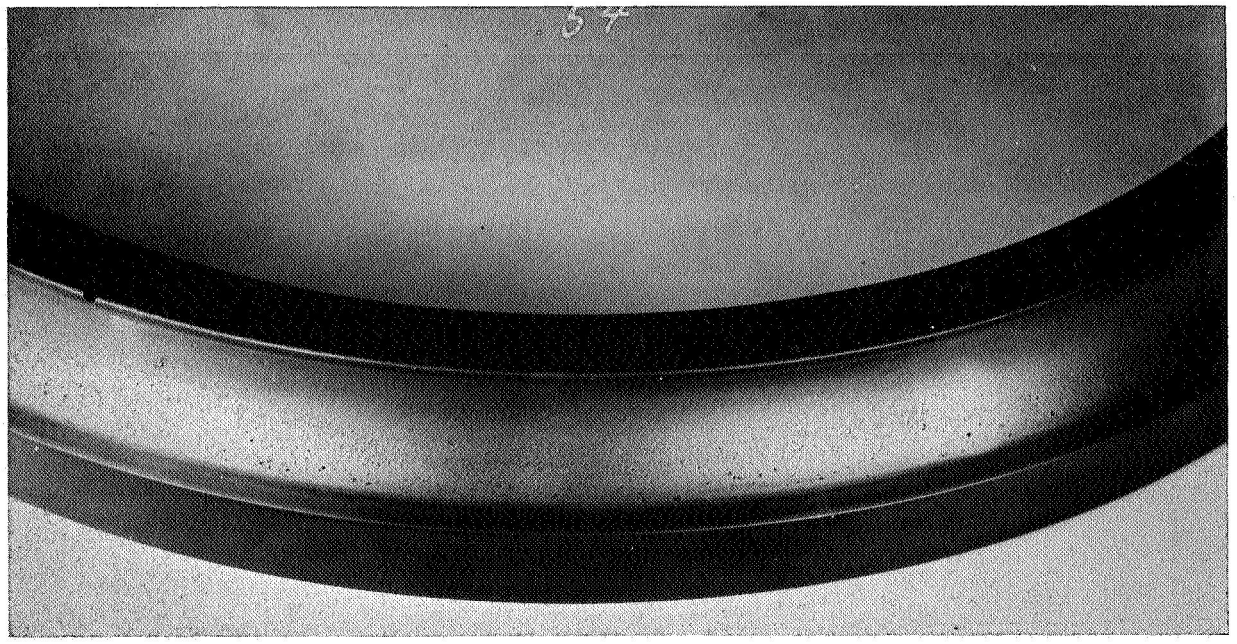


FIGURE 58. CAGE FAILURE AND BALL POCKET WEAR - BEARING S/N 54



FIGURF, 59. OUTER RING - BEARING S/N 54



FIGURF, 60. INNER RING - BEARING S/N 54

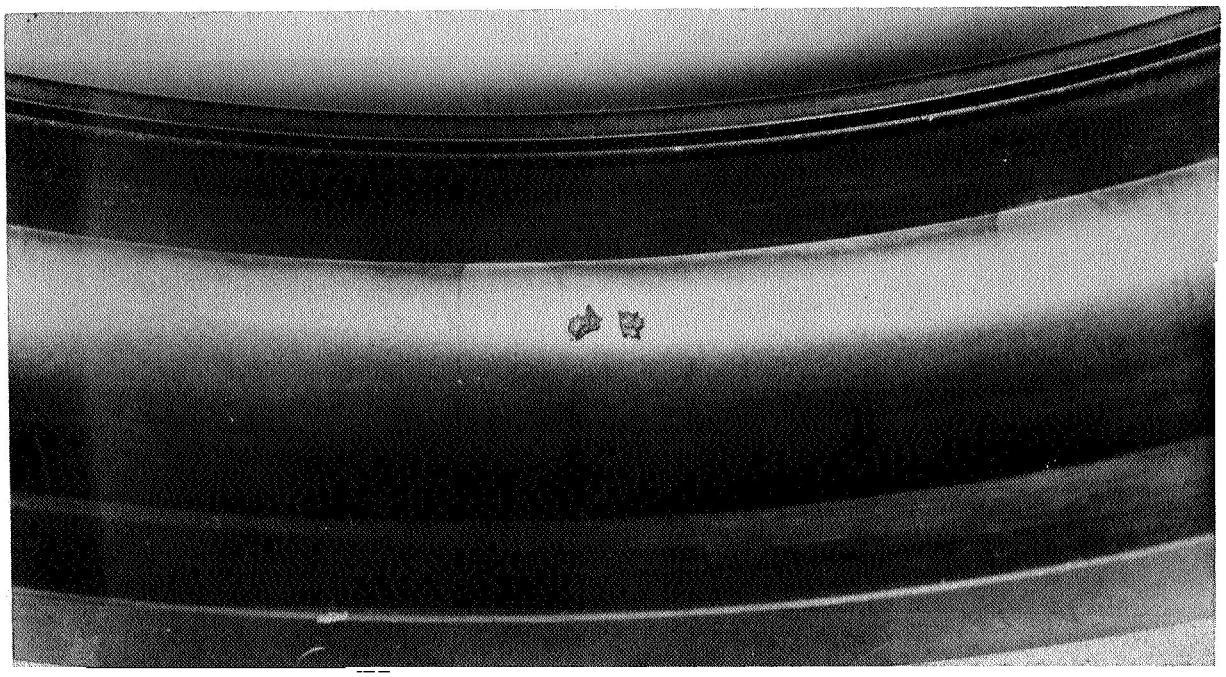


FIGURE 61. TYPICAL FAILURE IN BALL PATH IN OUTER RING - BEARING S/N 40

RUNNING TIME:	'45 MINUTES
LUBRICANT:	MCS-354 N_2 INERTED .
TEMPERATURE:	APPROXIMATELY 600F
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)

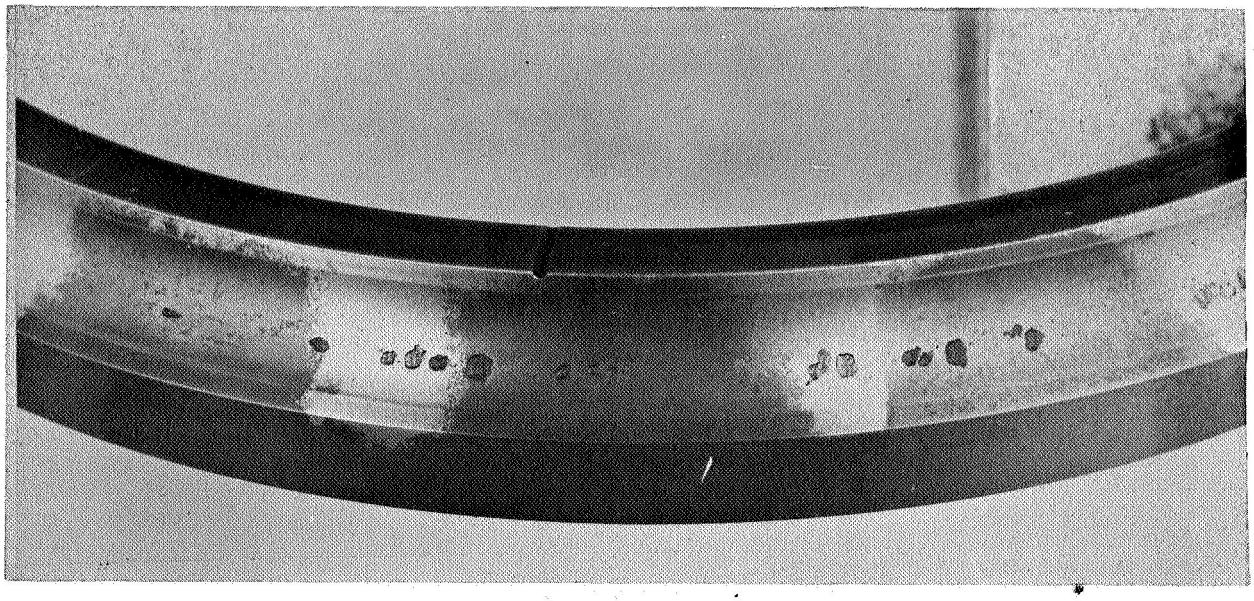


FIGURE 62. TYPICAL FAILURES ON BALL PATH OF INNER RING - BEARING S/N 40

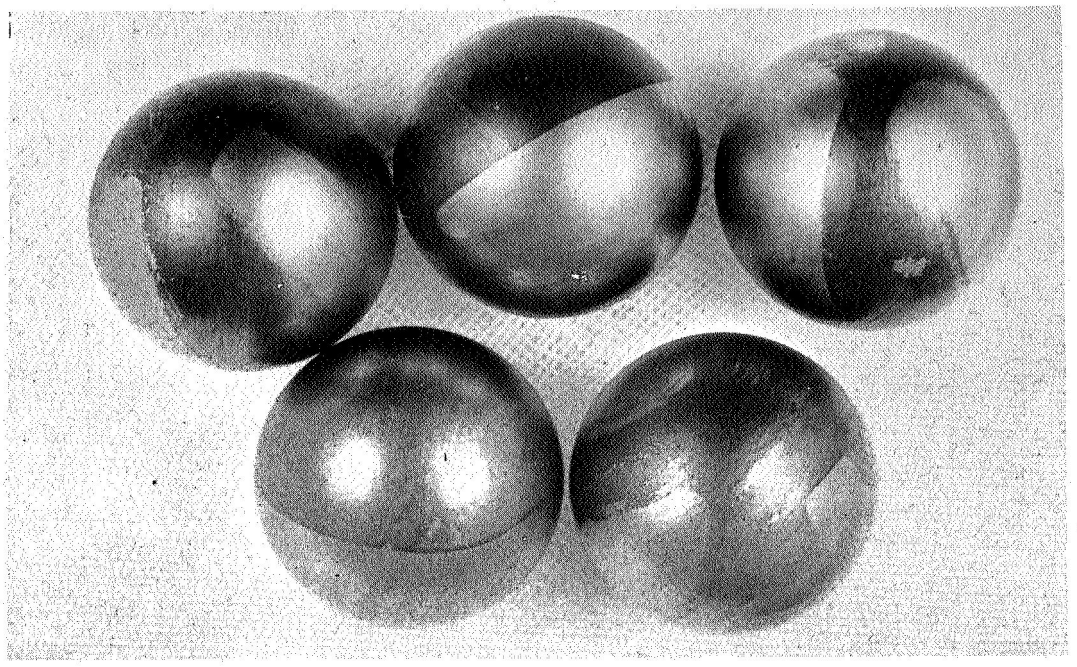


FIGURE 63. BALLS FROM BEARING S/N 40

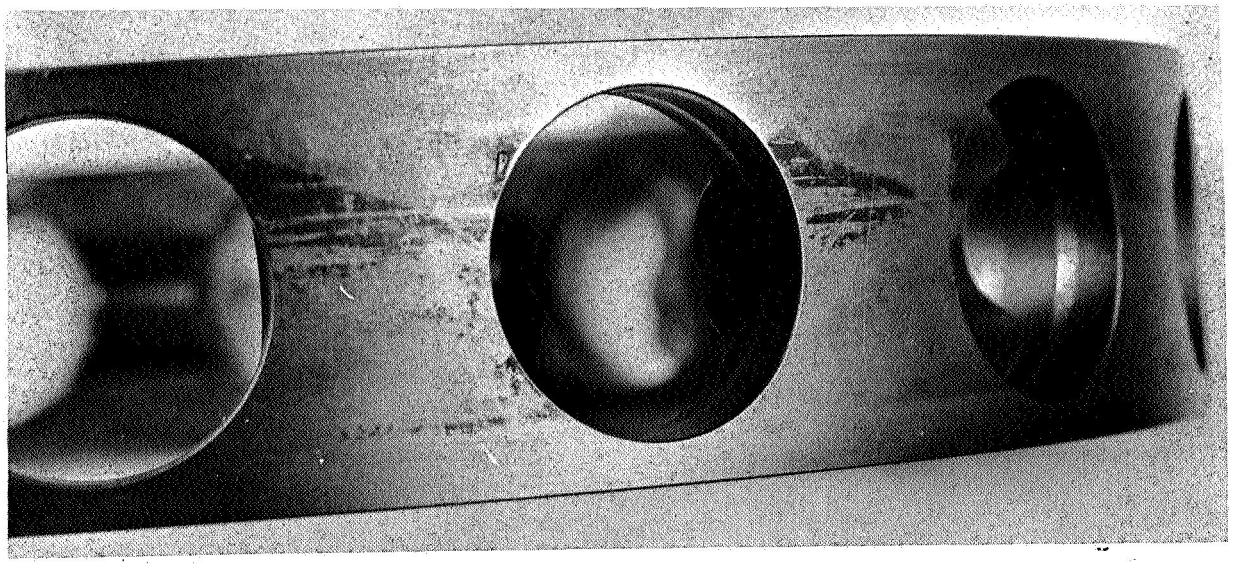


FIGURE 64. CAGE FROM BEARING S/N 40

Based on these results, the situation was again reviewed with NASA Project Management, and an agreement was reached to attempt to operate one set of bearings without the benefit of the nitrogen inertion. To accomplish this the N_2 line was disconnected and replaced with an air line. This was required, as a positive pressure had to be maintained in the tester in order for the J79 tandem carbon seal to be effective. A test was then conducted, running two M-50 bearings at 600F in an air atmosphere. This test continued successfully to 120 hours, at which point the bearing was disassembled and inspected. The bearing components appeared to be in good condition with no apparent evidence of wear or fatigue damage, These observations were reviewed with NASA Project Management, and the decision was made to conduct the entire second test series of this program in an air atmosphere.

Consequently, Test Series II was conducted under the following conditions:

Bearing Material;	Rings	CVM-M-50
	Balls:	CVM-M-50
	Retainer:	S-Monel
Lubricant:	Monsanto MCS-354	
Operating Conditions:		
	Temp:	Outer Ring 585F ± 5F
		Inner Ring 600F - 610F
		Oil In - 525F-545F
		Oil Out - 585F-595F
	Speed:	12,000 rpm
	Load:	4365 lbs, axial
	Atmosphere:	Air

Despite the reduced load, most of the tests with this fluid had to be aborted because of high ball wear. This is shown in the test results tabulated in Table 13, This data is also plotted in Weibull distribution in Figure 65, although the small number of actual fatigue failures (3 of 20)

Table 13

Results of 120 mm Bearing Tests with MCS-354 Lubricant, CVM M-50 Bearings, at 600F in Air

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
II - 1	53	5800 (2) (N_2)	2	OK	OK	OK (.003" Wear)
II - 1a	54	5800 (N_2)	2	OK	OK	OK (.002" Wear)
II - 2	39	4365 (1) (N_2)	.75	Fatigue	OK	OK (.001" Wear)
II - 2a	40	4365 (N_2)	.75	Fatigue	OK	OK (.001" Wear)
II - 3	45	4365	500	OK	OK	OK (.0002" Wear)
II - 3a	46	4365	500	OK	OK	OK (.0003" Wear)
II - 4	52	4365	14	OK	OK	Fatigue (.025" Wear)
II - 4a	51	4365	14	Fatigue	Fatigue	Fatigue
II - 5	47	4365	11	OK	OK	OK (.001" Wear)
II - 5a	48	4365	11	OK	OK	OK (.001" Wear)
II - 6	49	4365	65	OK	OK	OK (.0001" Wear)
II - 6a	50	4365	65	OK	OK	OK (.0002" Wear)
II - 7	61	4365	10	OK	OK	OK (.001" Wear)
II - 7a	62	4365	10	OK	OK	OK (.001" Wear)
II - 8	63	4365	20	OK	OK	OK (.001" Wear)
II - 8a	64	4365	20	Wear	Wear	OK (.025" Wear)
II - 9	67	4365	25	OK	OK	OK (.0005" Wear)
II - 9a	68	4365	25	OK	OK	OK (.001" Wear)
II -10	43	4365	500	OK	OK	OK (.0003" Wear)
II -10a	44	4365	500	OK	OK	OK (.0003" Wear)
II -11	57	4365	500	OK	OK	OK
II -11a	58	4365	500	OK	OK	OK

Table 13, Contd

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
II -12	59	4365	500	OK	OK	OK (.0002" Wear)
II -12a	60	4365	500	OK	OK	OK (.0002" Wear)
II -13	65	4365	269	OK	OK	OK (.0002" Wear)
II -13a	66	4365	269	OK	OK	OK (.0004" Wear)
II -14	69	4365	393	OK	OK	OK (.0002" Wear)
II -14a	70	4365	393	OK	OK	OK (.0005" Wear)
II -15	56	4365	461	OK	OK	OK (.0002" Wear)
II -15a	55	4365	461	OK	O	OK (.0002" Wear)

(1) 4365 lb. axial load = 295,000 psi max. hertz on inner.

(2) 5800 lb. axial load = 321,000 psi max. hertz on inner.

☐ Indicates Time to Fatigue Failure

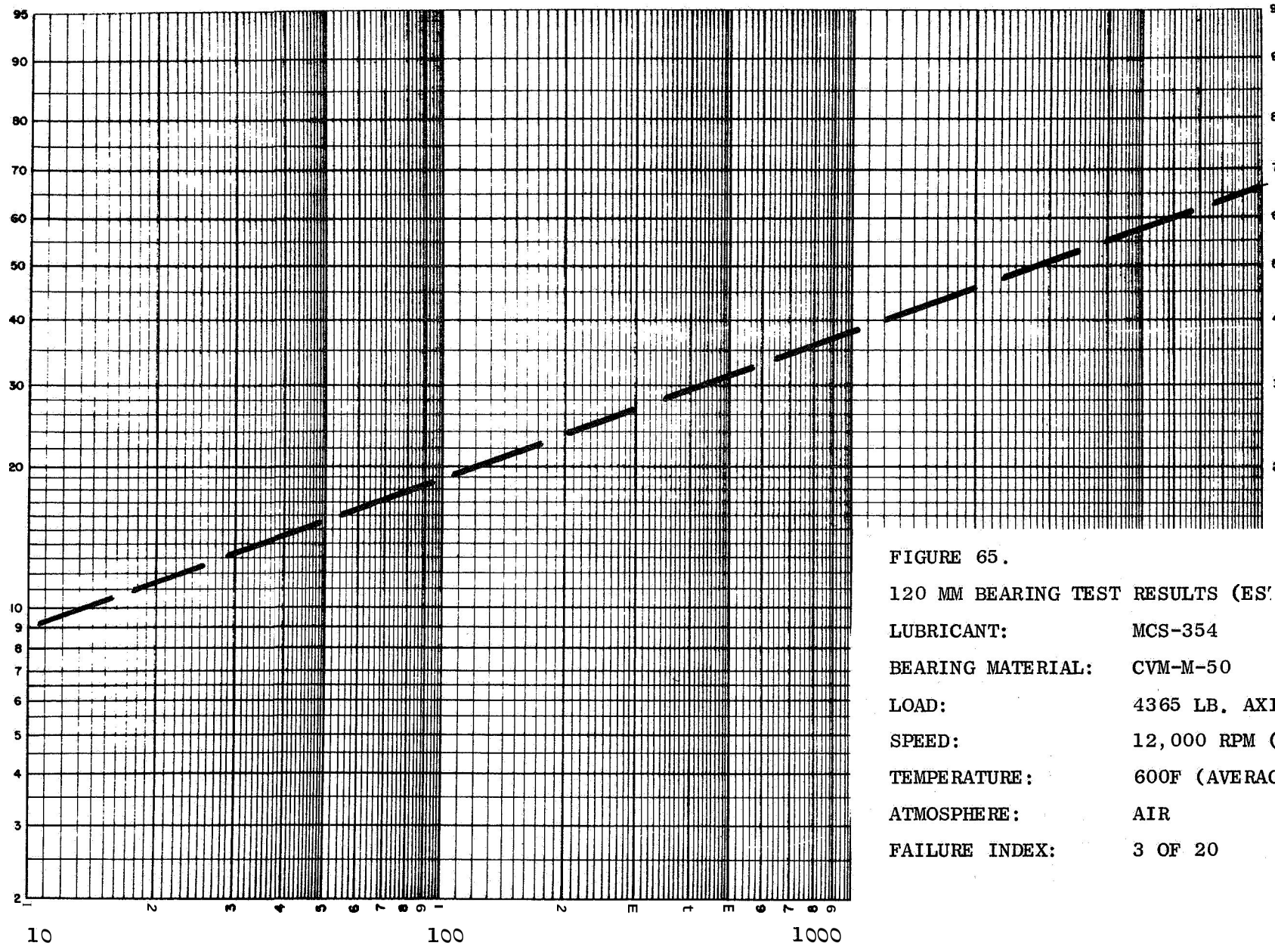


FIGURE 65.

120 MM BEARING TEST RESULTS (ESTIMATED)

LUBRICANT:	MCS-354
BEARING MATERIAL:	CVM-M-50
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	600F (AVERAGE METAL)
ATMOSPHERE:	AIR
FAILURE INDEX:	3 OF 20

carry a very low confidence level and tend to make this curve academic at best.

A significant result of the polyphenyl ether tests was the amount of ball wear observed on all of the bearings, even those which achieved the 500 hour life. In some bearings, ball wear was as high as .025", and the average wear in test times ranging from two to 65 hours was about .001". It was noted that in the case of the 500-hour bearings, ball wear while present, was relatively light, being on the order of .0003" or less, There does not appear to be an apparent relationship between wear and test time, and it can only be assumed that the variations could possibly be due to slight manufacturing tolerances which are not discernable in the allowable tolerance ranges; or it could be due to very small temperature differences such as ± 5-10F from one test to another. There also appears to be no clear-cut relationship between the initiation of wear and the rate at which wear proceeds.

Bearing 45 achieved 500 hours of running with little ball wear. Typical bearing components are illustrated in Figures 66 through 69. Of particular interest here is the surface finish on the balls (Figure 67), which is still quite good and devoid of micro pitting, despite the fact that ball wear ranging up to .0003" was measured. This was generally the case in that despite a considerable amount of wear on some of the balls, the surface finish remained good in contrast to the surface finish observed on the worn, XRM-177F lubricated balls. In the latter case, and in the case of the polyphenyl ether test performed in a nitrogen atmosphere, a considerable amount of micro pitting was present. However, this was not the case with the polyphenyl ether tested in air.

Several photographs are presented which are typical of some of the conditions observed on the bearings tested with the polyphenyl ether. Figures 70 and 71 are the balls and inner raceway of

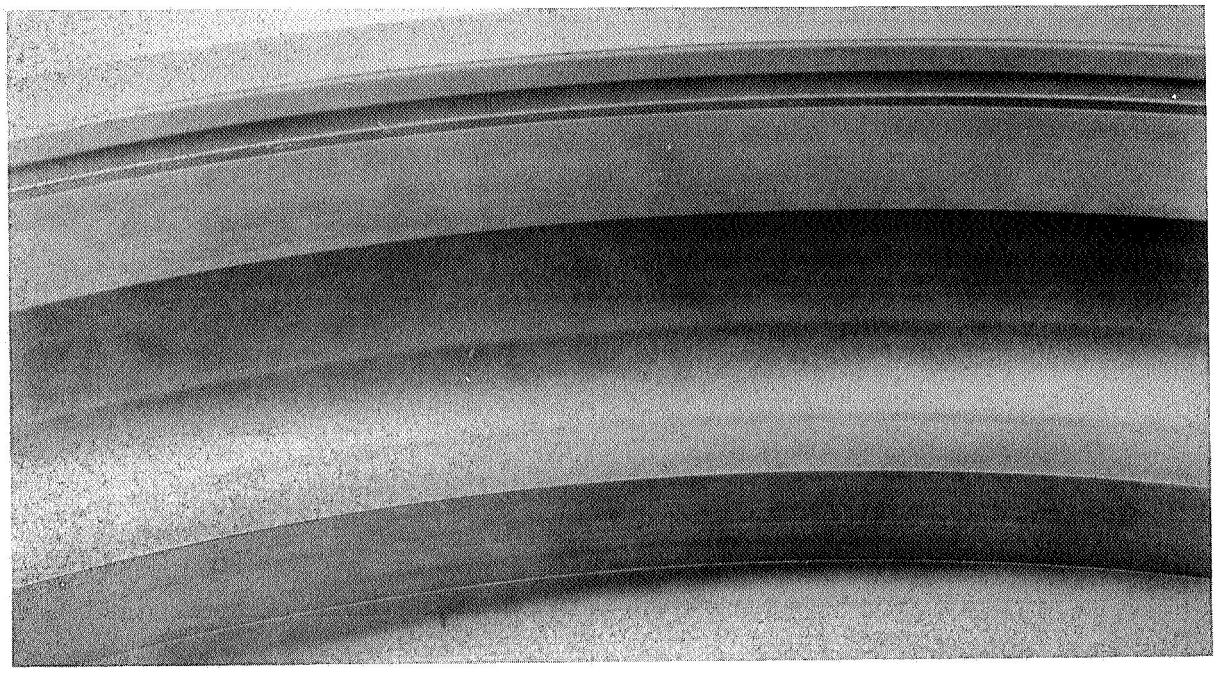


FIGURE 66. OUTER RING BALL TRACK - BEARING S/N 45

RUNNING TIME:	500 HOURS
LUBRICANT:	MCS-354 (AIR ATMOSPHERE)
TEMPERATURE:	600F
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)

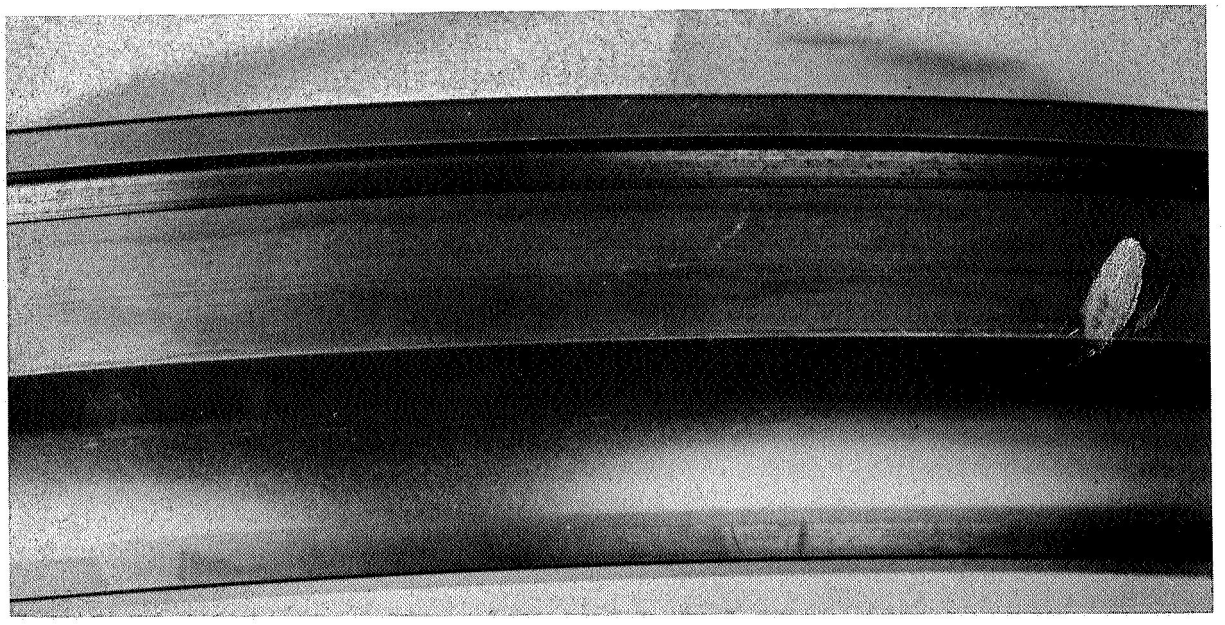


FIGURE 67. INNER RING BALL PATH - BEARING S/N 45

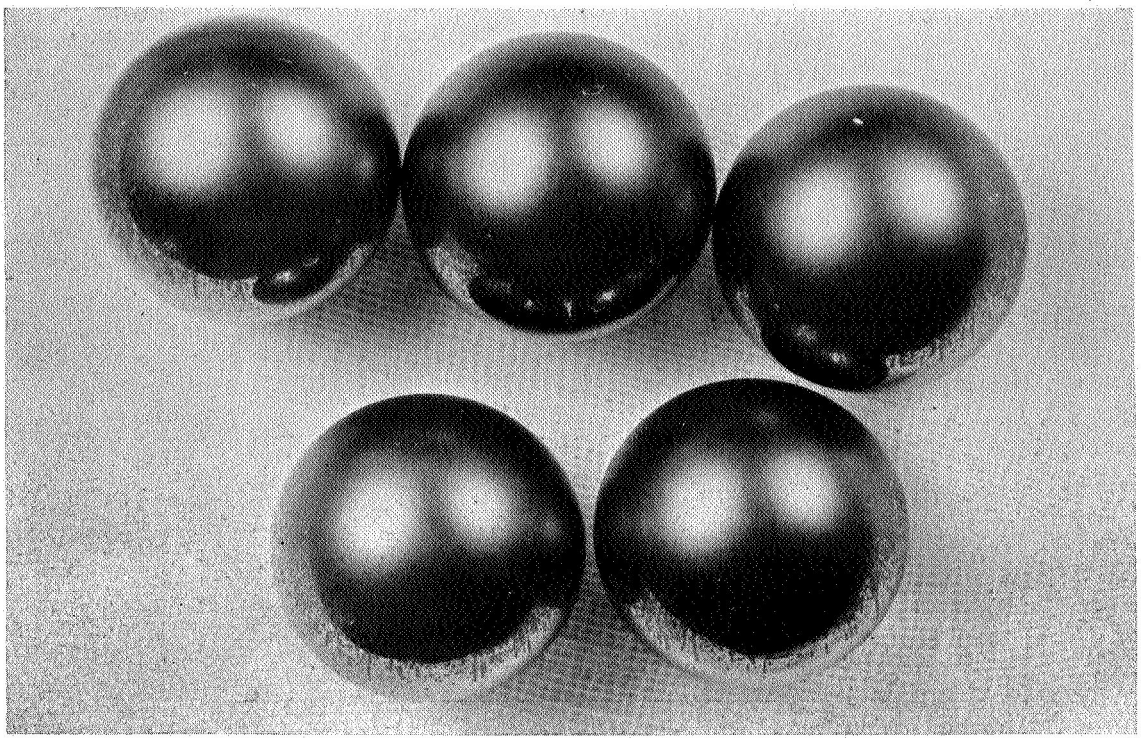


FIGURE 68. CAGE BALL POCKET - BEARING S/N 45

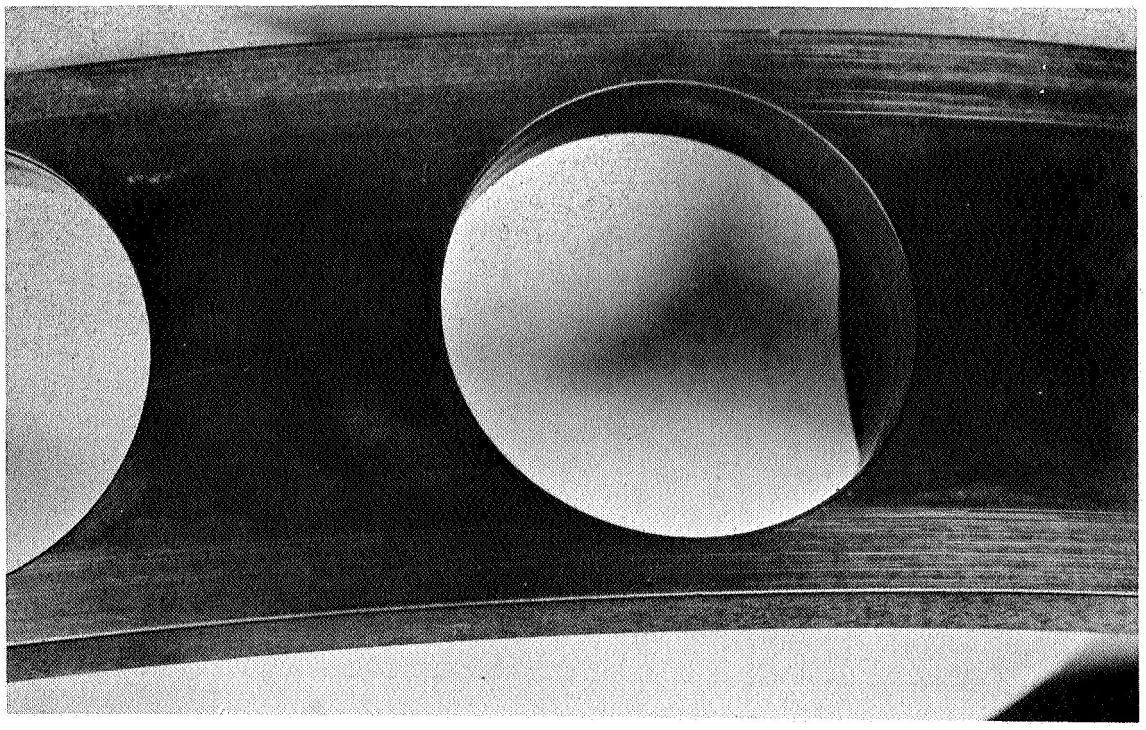


FIGURE 69. BALLS - BEARING S/N 45

bearing 52, which operated for 14 hours at the 4365 lb. level and showed excessive ball wear (.025") as well as ball fatigue. The unusual pattern shown op the balls in Figure 70 is due to staining after shut-down rather than to actual operation. The next four photographs are illustrative of the condition found in bearing 51, which also had accumulated 14 hours. Figure 72 shows the fatigue spalls on several of the balls, and it is interesting to observe the somewhat unusual texture of the failure. The failures appear to be much mare brittle than normally observed with compressive fatigue failures (where some evidence of plastic deformation is almost always evident). Figure 73 shows a cage side rail fracture and the damage caused by rubbing against the outer ring after the cage fractured. Figures 74 and 75 are typical fatigue spalls noted on the inner and outer raceways. Note also the damage on the raceway shoulder which occurred when, the ball diameter was reduced sufficiently to allow a significant contact angle increase, thereby causing the ball to ride up over the shoulder of the raceway.

With the polyphenyl ether, oil consumption was quite high. Attempts at refluxing were unsuccessful, as very little oil can be reclaimed using this technique. It appears that the basic problem in oil loss is the relatively high evaporation rate of this material. Based on data presented by Monsanto,[9] the evaporation loss as measured by ASTM method D-972 is 8 1/2%, in 6 1/2 hours at 500F at 140 millimeters mercury of pressure. This would indicate that in a 24-hour period, approximately 30% of the fluid would be lost due to evapsration at a temperature lower than that to which the fluid was exposed in this program.

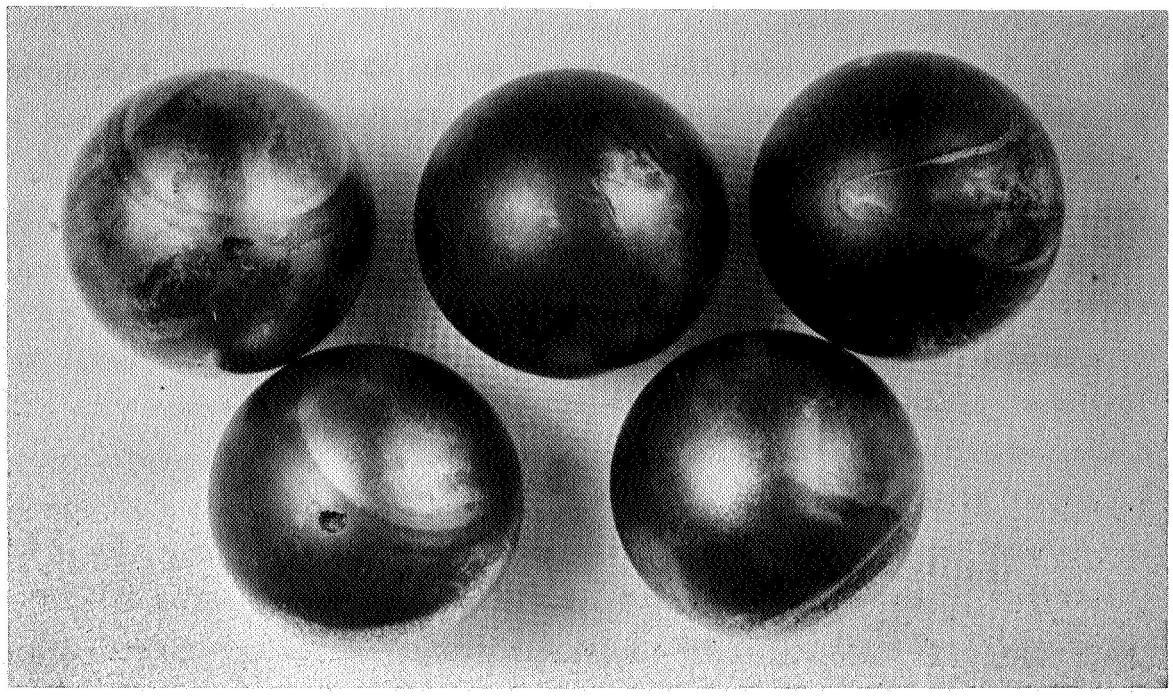


FIGURE 70. BALL FATIGUE FAILURES - BEARING S/N 52

RUNNING TIME:	14 HOURS
LUBRICANT:	MCS-354 (AIR ATMOSPHERE)
TEMPERATURE:	600F
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)

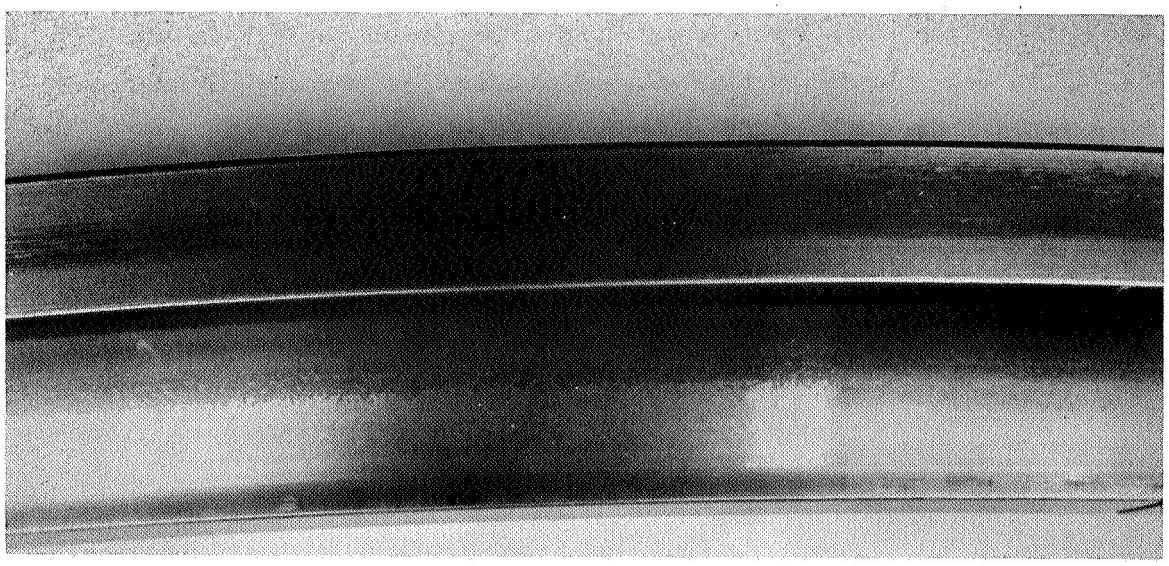


FIGURE 71. INNER RING BALL PATH - BEARING S/N 52

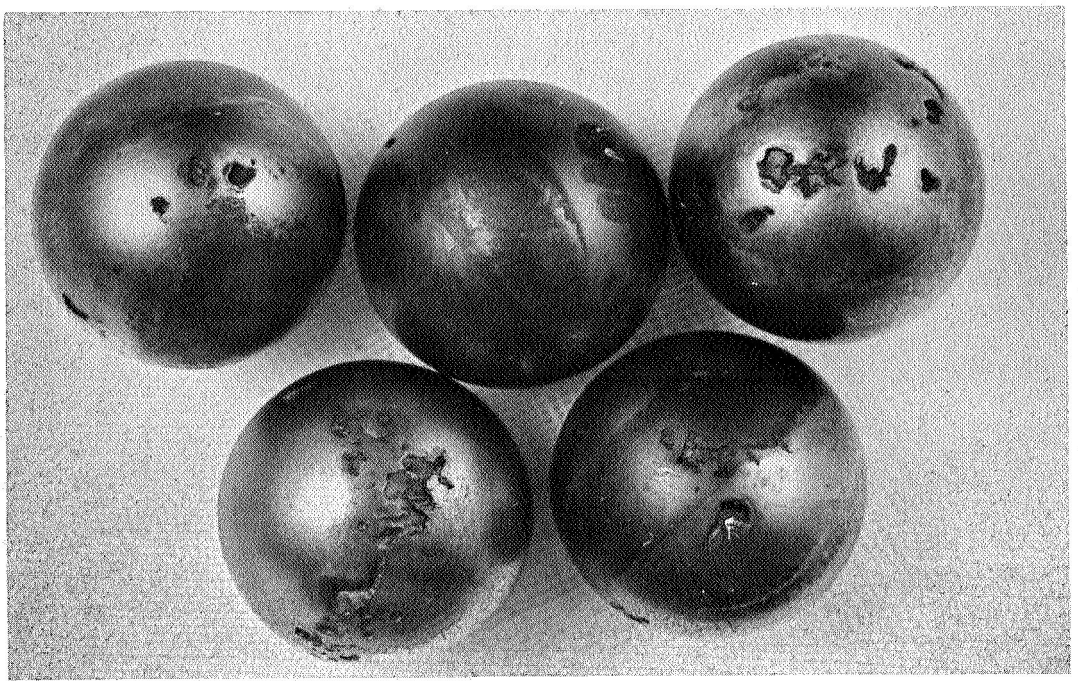


FIGURE 72. BALL FATIGUE FAILURES - BEARING S/N 51

RUNNING TIME:	14 HOURS
LUBRICANT:	MCS-354 (AIR ATMOSPHERE)
TEMPERATURE:	600F
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)

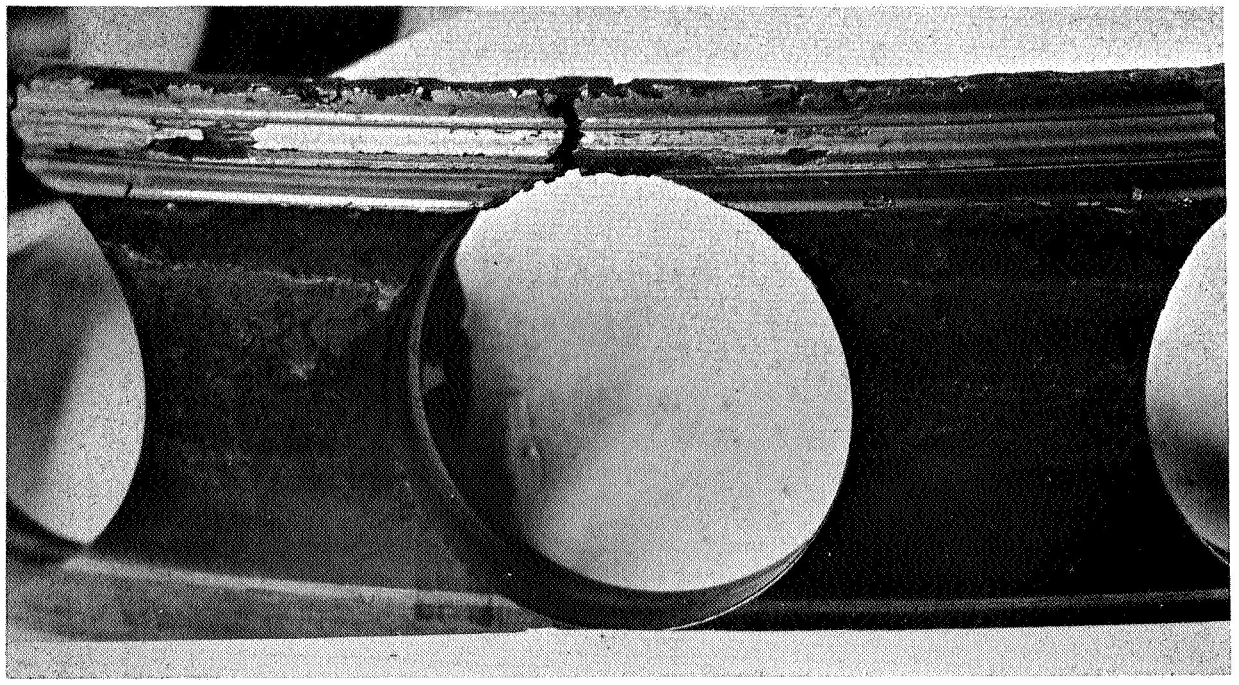
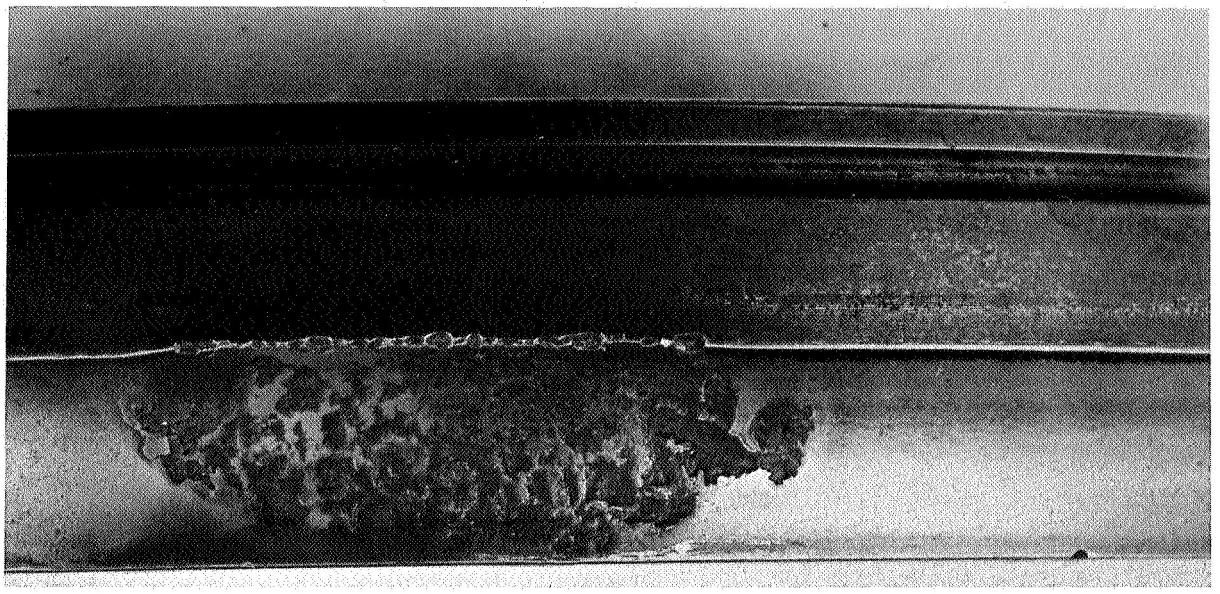
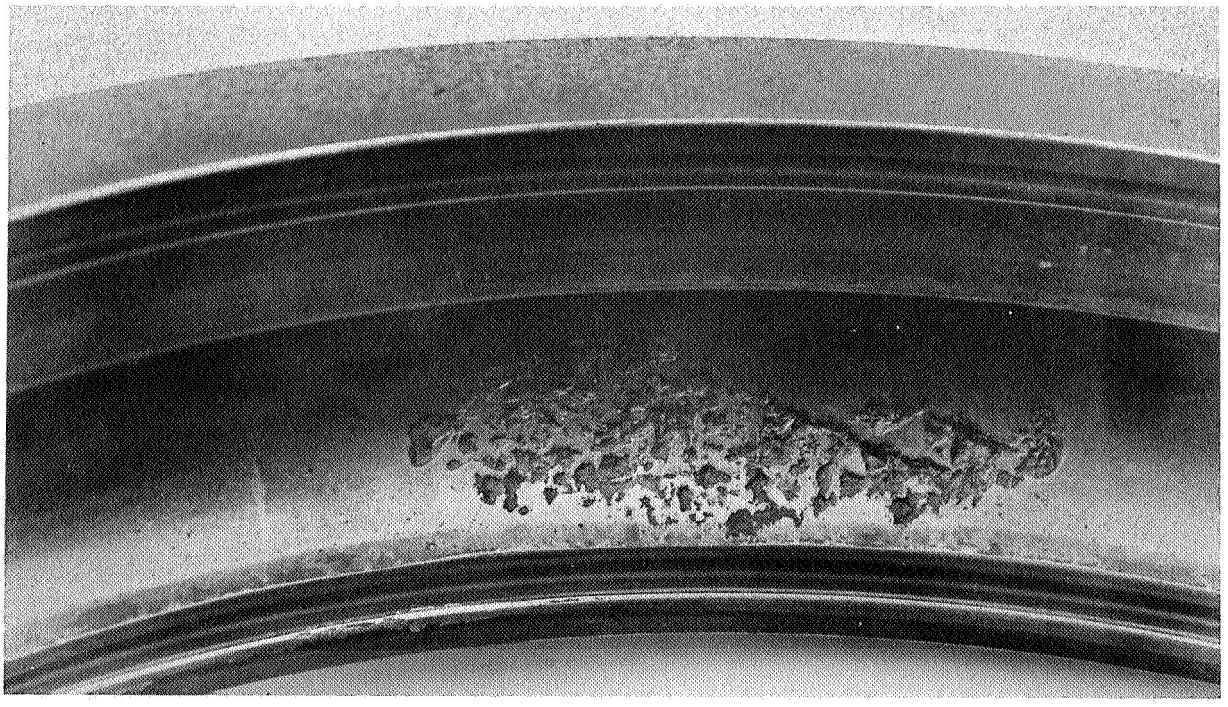


FIGURE 73. CAGE FAILURE - BEARING S/N 51



FIGURJI 74. INNER RING FATIGUE FAILURE - BEARING S/N 51



FIGURJI 75. OUTER RING FATIGUE FAILURE - BEARING S/N 51

6.2.3 Test Series III

Prior to the initiation of this test series, all tester components were inspected for the quality of nickel plating and those components showing evidence of plating damage were replated.

Again, as with the two earlier test series it had been planned to operate at 5800 lbs. axial load and in a N_2 atmosphere. However, a number of problems were encountered which required a deviation in the program plan,

In the initial check-out test with the PR-143, it was Pound that the oil-water heat exchanger used to cool the oil, in the previous tests did not have sufficient capacity to maintain the desired temperature range of 585P to 610F. This is primarily due to the higher density of this fluid as well as its low specific heat and low thermal conductivity as compared to the other two test fluids. It was found that the temperature tended to approach stabilization at approximately 100° above the temperature at which the two previous fluids had stabilized. This is in agreement with data presented by other sources who have also performed Pull scale bearing testing wftb the DuPont PR-143 fluid,

To overcome this problem, a new heat exchanger was designed and checked out, In order to accomplish this in the least amount of time, it was necessary to deviate from the all Monel and nickel plated metallic components which had been judged necessary to provide protection against corrosive effects. Consequently, in the first series of tests using the new heat exchanger, stainless steel (316) fittings and Bundyweld tubing were utilized, The Bundyweld tubing is a low carbon steel tubing encased in a thin layer of copper, Using two M-50 bearings, specifically released by NASA for this study, a 100-hour test was conducted at an axial load of 5800 lbs., a speed of 12,000 RPM, and a bearing temperature of 600F. Samples of the ail were inspected at frequent periods during this run. Initially, the oil assumed a milky color, although after about 50 hours of running, the coloration of the oil became a very light brown, remaining

this color for the 100 hour period, Spectographic analysis of the oil after 100 hours showed only very minor traces of metallic elements. After 100 hours the tester was disassembled and samples of the stainless steel fittings as well as the Bundyweld tubing were examined metallographically. The stainless steel showed no evidence of corrosion, there was, however, some indication of copper depletion on the ID of the Bundyweld tubing indicating this material to be marginal for this particular application. Based on these results, it was decided that the new heat exchanger system would be constructed entirely of 316 stainless steel components. The lack of corrosion on stainless steel is understandable, in that the maximum oil temperature to which this material is exposed to is in the range of 500F to 525F. The surfaces of the tester per se appeared to be unaffected, although all surfaces which had been in contact with the oil were noticeably darkened. This, of course, included the test bearings.

The test bearings appeared to be in good condition except for some ball wear. Figures 76 through 78 illustrate typical components after the 100 hour test. It was decided however, that in view of the possible contaminant effect of the Bundyweld tubing and copper that these bearings would be shelved.

In all other respects the behavior of the oil was normal and showed evidence of performance consistent with its high density and low specific heat. The viscosity at 600F is similar to the other two fluids in the program. Because of the high density, an increased pressure was developed in the centrifugal pump since pressure varies as:

$$\Delta P = r^2 W^2 \rho$$

where ΔP = pressure in psia

ρ = fluid mass density per in^3

r = radius in inches

W = rotational speed in rad. per sec.

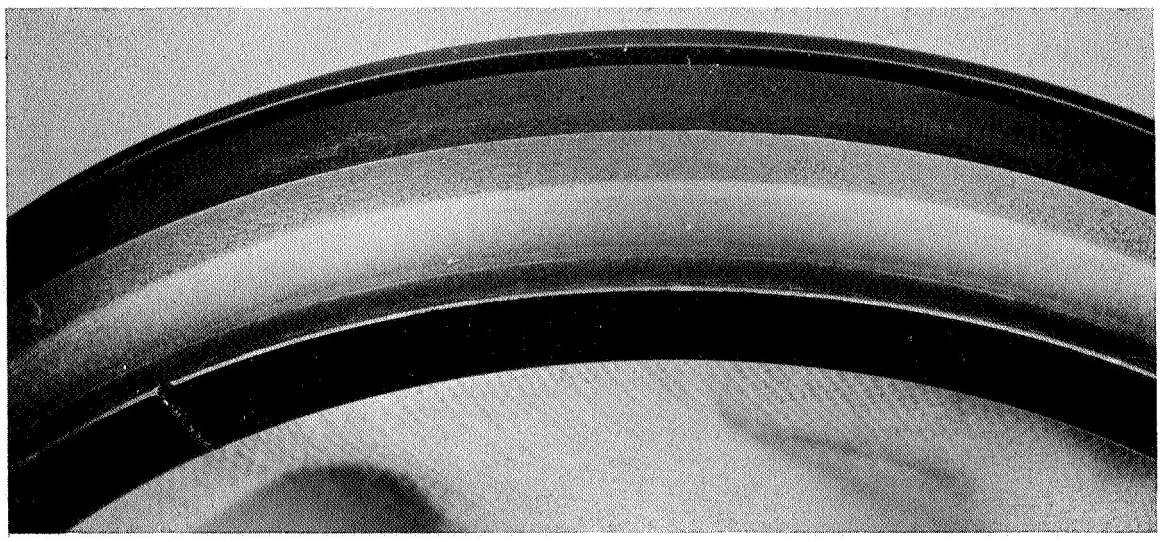
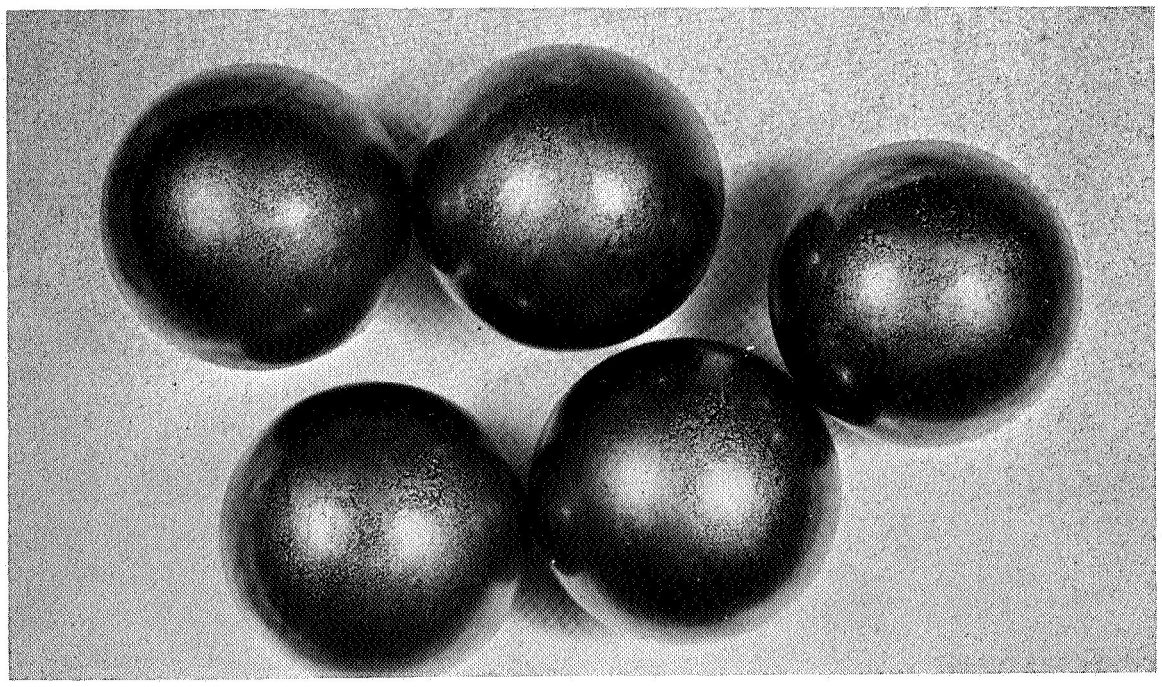


FIGURE 76. BALL TRACK ON INNER RING OF BEARING S/N 71 AFTER 100 HOUR$

TEST CONDITIONS:

FLUID:	PR-143
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F



TYPICAL APPEARANCE OF BALLS FROM BEARING S/N 71 AFTER 100 HOURS

NOTE: THE MOTTLED APPEARANCE OF THE BALLS IS *AN* ARTIFACT, CAUSED BY THE DULLING SPRAY PUT ON THE PARTS PRIOR TO PHOTOGRAPHY.

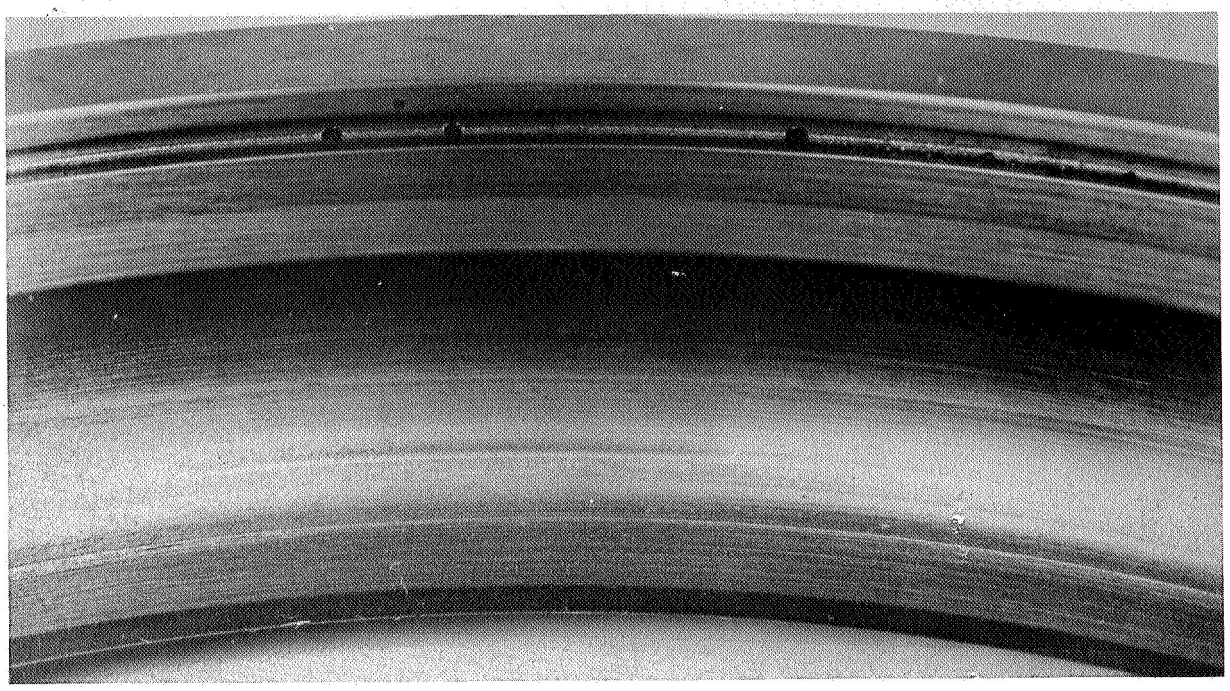


FIGURE 77. BALL TRACK ON OUTER RING OF BEARING S/N 71 AFTER 100 HOURS.

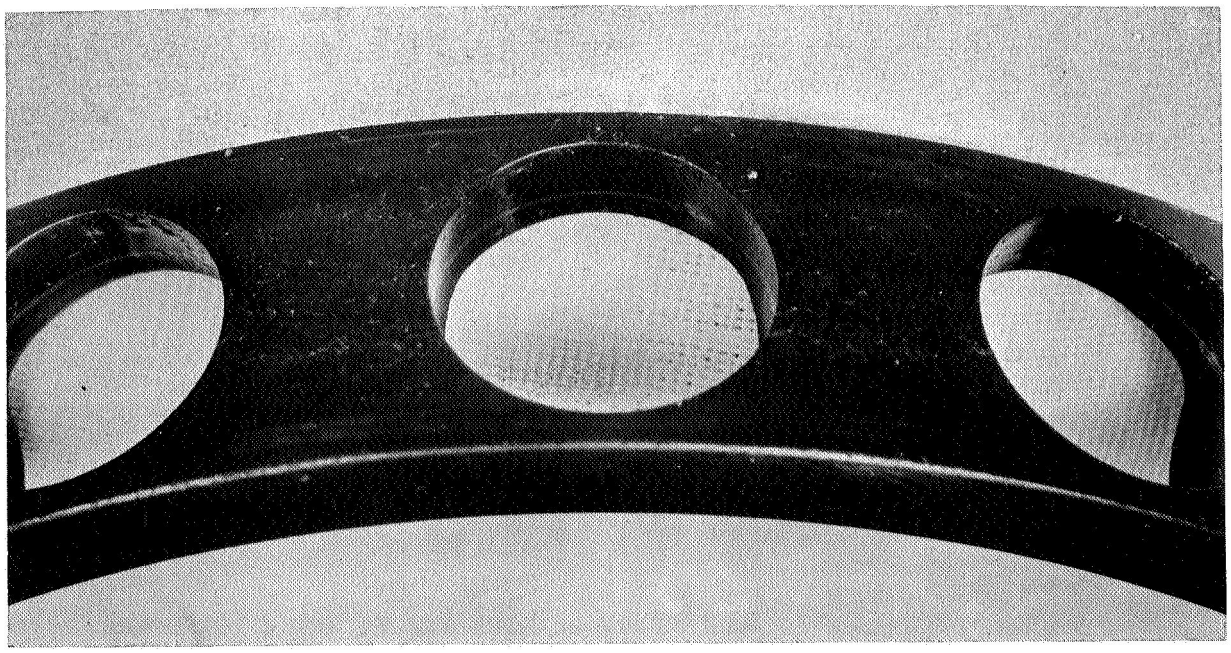


FIGURE 78. RETAINER FROM BEARING S/N 71 AFTER 100 HOURS.

In addition, an increased power loss was observed in the bearing due to the same inertial energy effects.

The foaming problems encountered previously with the other two fluids were not present. However, the high volatility of the fluid necessitated the addition of about 1 1/2 gallons of make up fluid during the 100 hours of teats. This is approximately 50% of the initial charge of oil,

Testing was resumed at the 5800 lb. level, although considerable ball wear and/or surface distress was observed. Consequently, after 8 tests at this load, the testing was halted and the situation reviewed with NASA Project Personnel.

As a result of these discussions, it was again decided to reduce the load to that used for the MCS-354 (4365 lb, axial). The remaining tests in Series III were therefore performed at this reduced load level.

Test Series III therefore was conducted under the following conditions:

Bearing Material:	Rings:	CVM-M-50
	Balls:	CVM-M-50
	Retainer:	S-Monel
Eubricant:	DuPont PR-143	
Operating Conditions:		
	Temp:	Outer Ring 585F ± 5F
		Inner Ring 600F - 610F
		Oil In - 525F-560F
		Oil Out - 590F-615F
	Speed:	12,000 rpm
	Load:	5800 lbs. axial (8 tests)
		4365 lbs. axial (22 tests)

The data from these tests is tabulated in Table 14.

Table 14

Results of 120 mm Bearing Tests with PR-143 Lubricant, CVM M-50 Bearings, @ 800F in N_2

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
IIII-1	77	5800 (1)	103	OK	OK	OK (.0003" Wear)
IIII-1a	78	5800	103	Damaged (2)	Damaged	Damaged
IIII-2	75	5800	243	OK	OK	OK (.0004" Wear)
IIII-2a	76	5800	243	OK	OK	OK (.0004" Wear)
IIII-3	79	5800	201	OK	OK	OK (.0015" Wear)
IIII-3a	80	5800	201	Damaged	Damaged	Damaged
IIII-4	83	5800	202	OK	OK	OK (.0005" Wear)
IIII-4a	84	5800	202	Damaged	Damaged	OK (.0005" Wear)
IIII-5	85	4365 (3)	171	OK	OK	OK (.0005" Wear)
IIII-5a	86	4365	171	Fatigue	OK	Fatigue
IIII-6	81	5800	506	OK	Fatigue	OK (.0020" Wear)
IIII-6a	82	5800	506	OK	OK	OK (.0020" Wear)
IIII-7	73	5800	436	Fatigue	Fatigue	OK (.0025" Wear)
IIII-7a	74	5800	436	Fatigue	Fatigue	OK (.0025" Wear)
IIII-8	93	4365	159	Damaged	Damaged	Fatigue
IIII-8a	94	4365	159	OK	OK	OK
IIII-9	87	4365	500	OK	OK	OK (.0002" Wear)
IIII-9a	88	4365	500	OK	OK	OK
IIII-10	89	4365	134	Damaged	Damaged	OK (.0005" Wear)
IIII-10a	90	4365	134	OK	OK	OK (.0005" Wear)
IIII-11	99	4365	.01	OK	OK	OK
IIII-11a	100	4365	.01			

Table 4 Cont'd

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
III -12	95	4365	91	OK	OK	OK (.0005" Wear)
III -12a	96	4365	91	Damaged	Damaged	OK (.0005" Wear)
III -13	97	4365	142	OK	Fatigue	OK (.0005" Wear)
III -13a	98	4365	142	OK	OK	OK (.0005" Wear)
III -14	91	4365	500	OK	OK	OK (.0004" Wear)
III -14a	92	4365	500	OK	OK	OK (.0005" Wear)
III -15	99	4365	403	OK	Fatigue	OK (.0015" Wear)
III -15a	101	4365	403	OK	OK	OK (.0015" Wear)

(1) 5800 lb. axial load = 321,000 psi max. hertz on inner.

(2) Damage refers to surface distress as a result of seizure.

(3) 4365 lb. axial load = 295,000 psi max. hertz on inner.

☐ Indicates Time to Fatigue Failure

The fatigue failures at the lower load are presented in a Weibull distribution in Figure 79.

During these tests, continued problems with high power loss due to high oil density and marginal boundary film conditions were encountered. Belt life, normally in excess of 500 hours, was reduced severely. Damage to shafts and pumps was also a problem and as a result, a number of the major tester components required repair and/or overhaul during and after this test series. Of the three oils tested, the PR-143 precipitated the most severe and recurrent operational problems. From an engine application standpoint, the most serious of these problems was the significantly greater difficulty in rejecting heat which required re-sizing the heat exchanger.

The photographs shown in Figures 80 through 97 are presented as typical examples of the appearance of the M-50 bearings tested with the PR-143. Figures 80 to 83 are views of Bearing S/N 78 which suffered a catastrophic failure after 103 hours. More normal fatigue failures are shown in Figures 84 and 85, whereas Figures 86 through 88 illustrate the appearance of a bearing (S/N 81) which successfully completed 500 hours of test. Figures 89 and 90 are examples of fatigue failure encountered after 436 hours of operation at the 5800 lb. axial load. Figure 89 shows a normal fatigue failure on the inner ring of bearing S/N 74, whereas Figure 90 shows a second type of failure encountered. This one being on the outer ring of the same bearing, The failures here are more similar to those experienced with the polyphenylether (MCS-354). Figure 91 is a close-up of a surface of the inner ring of bearing S/N 74 showing the type of chemical corrosion observed on the long life bearings, Figure 92 shows an example of surface damage encountered at the lower load level (4365 lbs.). The S-Monel retainer of this bearing (Figure 93) appears to be in excellent condition, as was the case with most of these components during this and the previous test series. Figures 94 through 96 are additional examples of bearing components having successfully survived 500 hours of testing. The last photograph (Figure 97) shows the cage rub experienced after .01 hours of operation on bearing S/N 100,

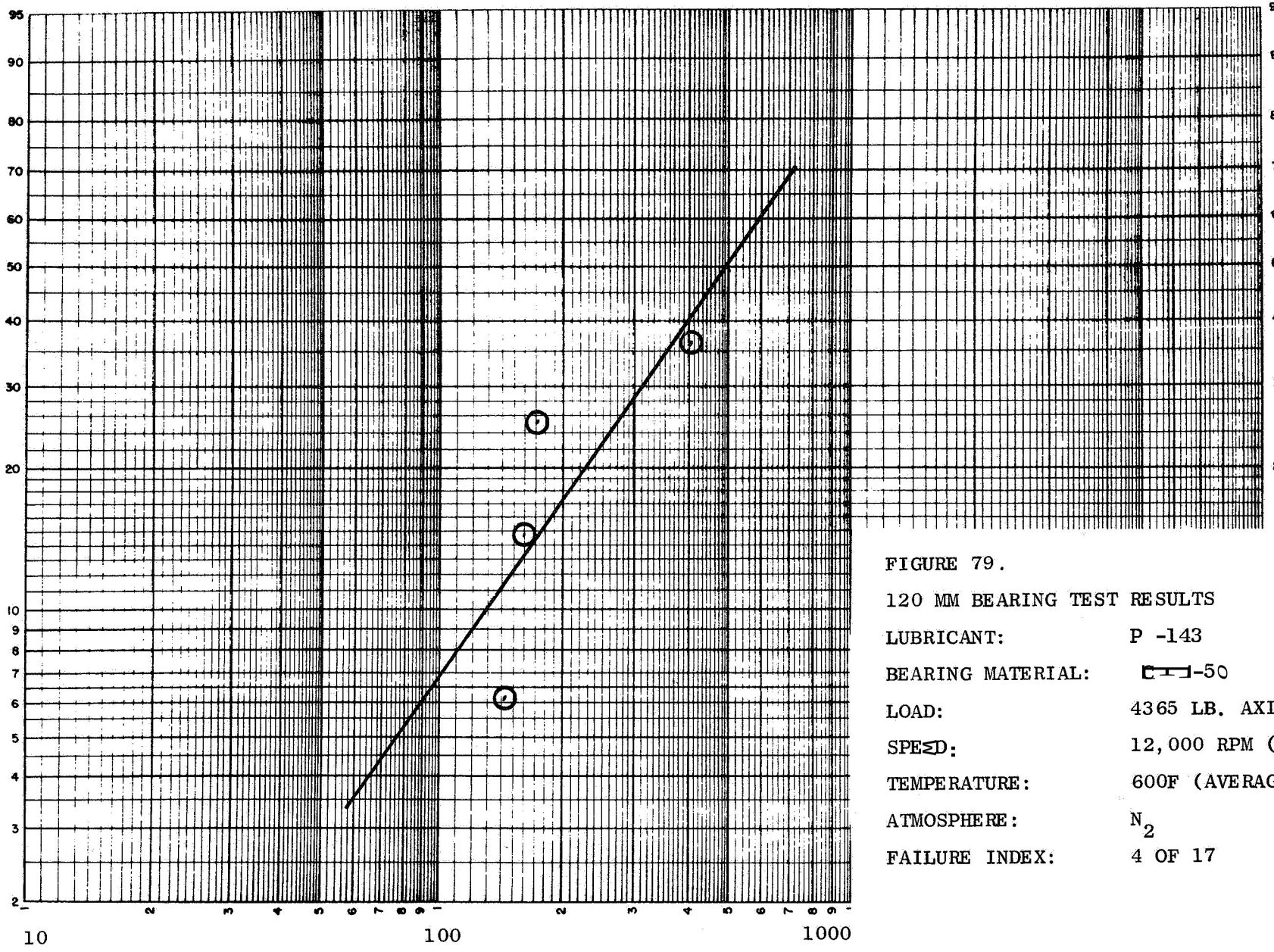


FIGURE 79.

120 MM BEARING TEST RESULTS

LUBRICANT:	P -143
BEARING MATERIAL:	C-50
LOAD:	4365 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	600F (AVERAGE METAL)
ATMOSPHERE:	N_2
FAILURE INDEX:	4 OF 17

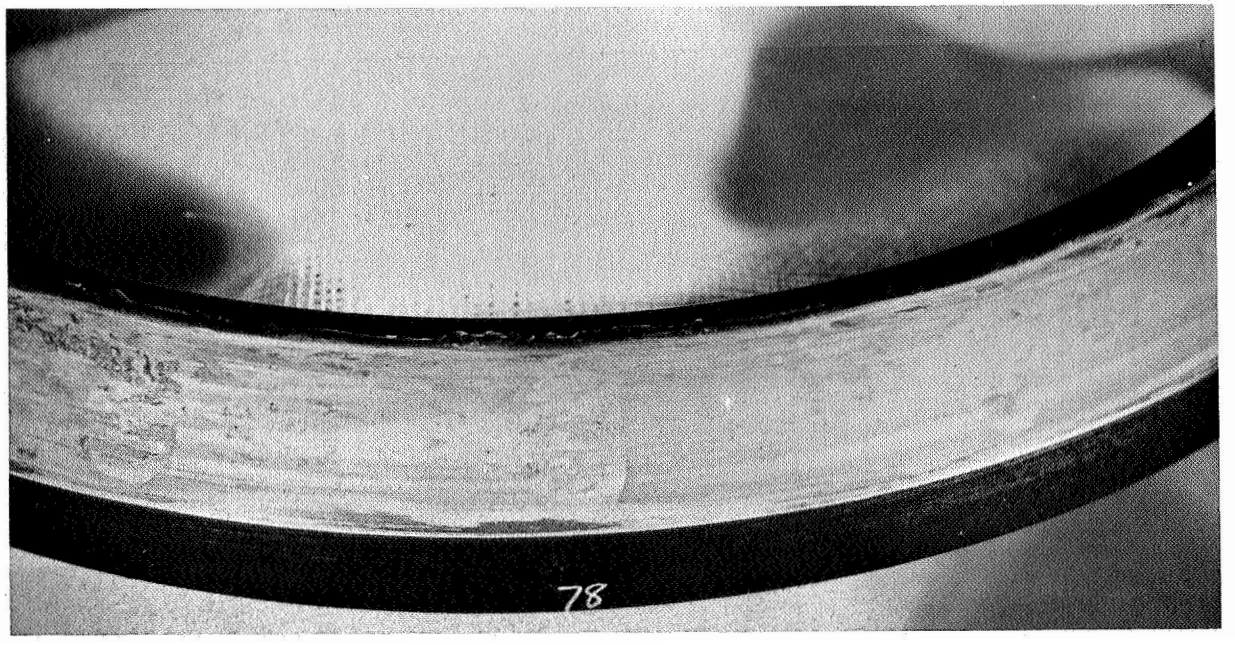


FIGURE 80. BEARING S/N 78. DAMAGED INNER RING.

TEST CONDITIONS:

LOAD:	5800 LB, AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	103 HOURS

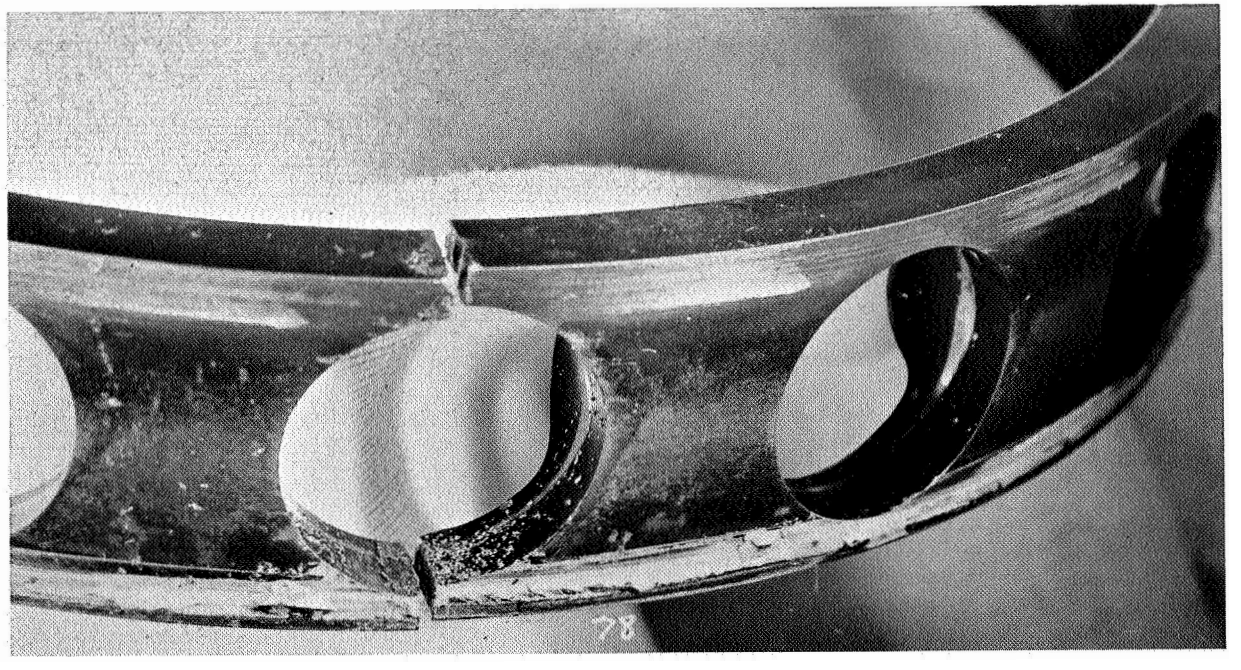


FIGURE 81. BEARING S/N 78. FRACTURED CAGE. NOTE EVIDENCE OF CAGE RUB.

FIGURF: 82. BEARING S/N 78. DAMAGED BALLS.

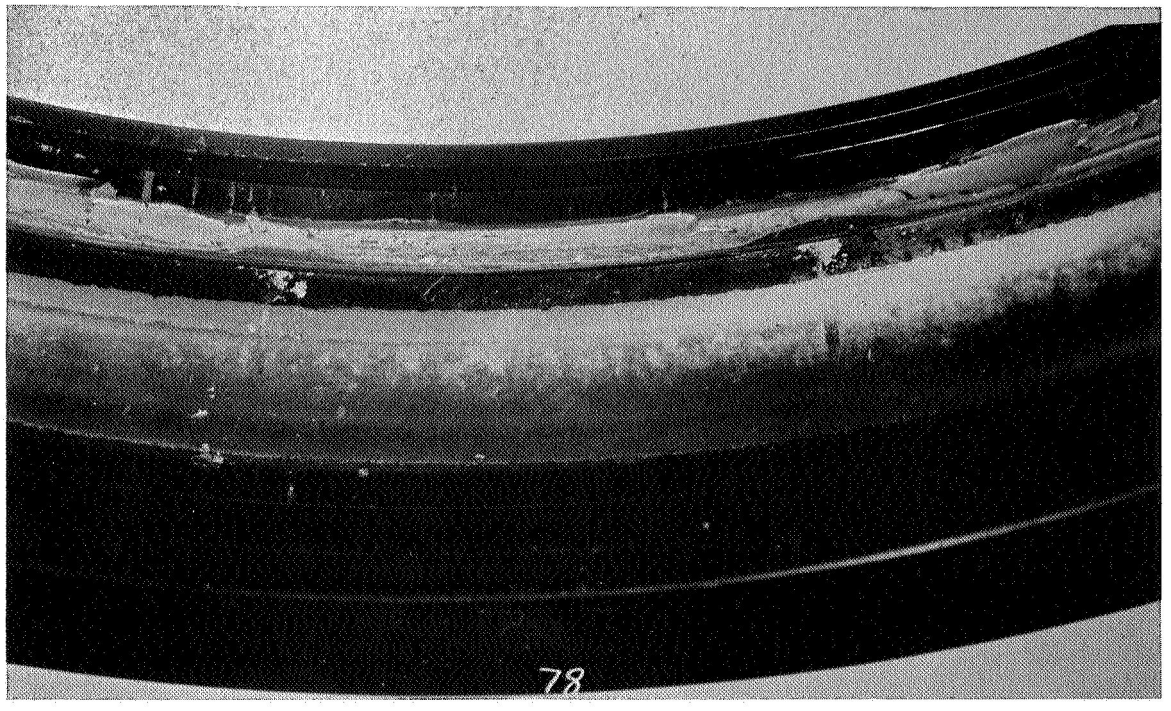


FIGURE 83. BEARING S/N 78. **DAMACED** OUTER RING.

FIGURE 84. BEARING S/N 73. FATIGUE FAILURE ON INNER RING.

TEST CONDITIONS:

LOAD:	5800 LBS.
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	436 HOURS

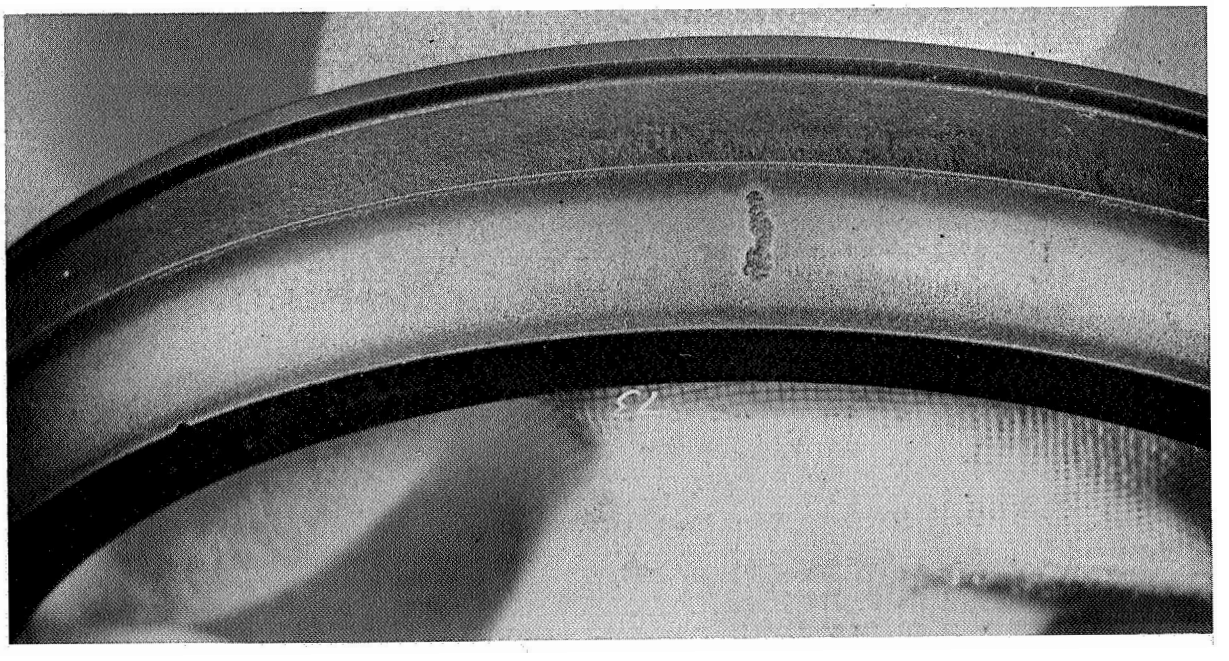


FIGURE 85. BEARING S/N 73. FATIGUE FAILURE ON OUTER RING.

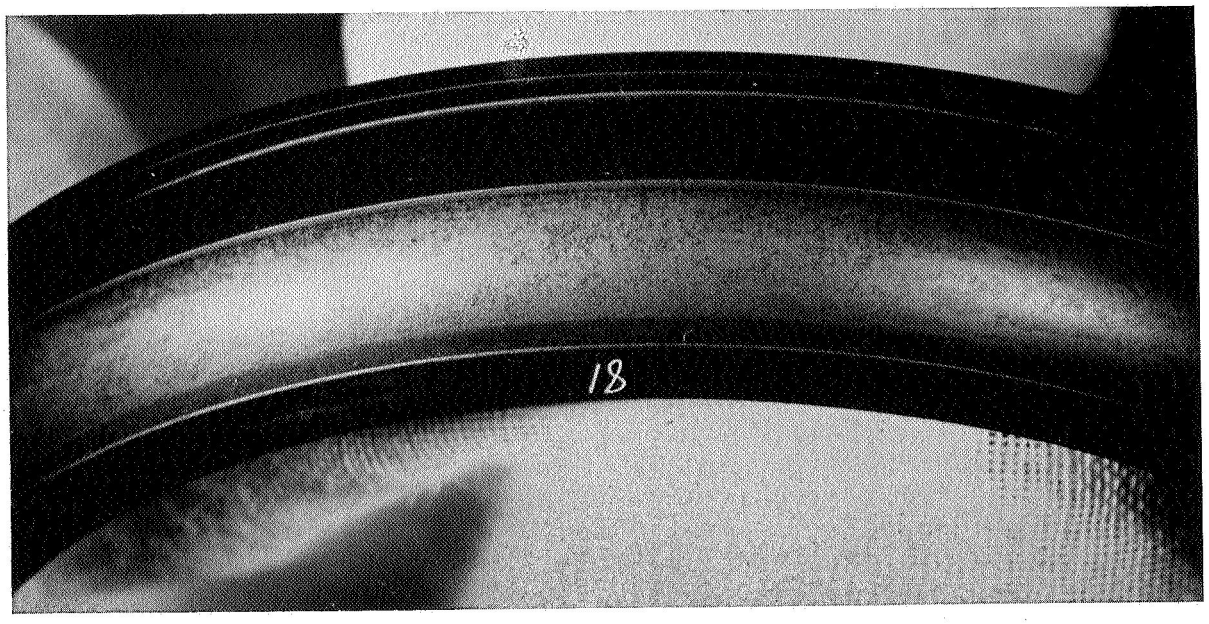


FIGURE 86. BEARING S/N 81. INNER RING.

TEST CONDITIONS:

LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	506 HOURS

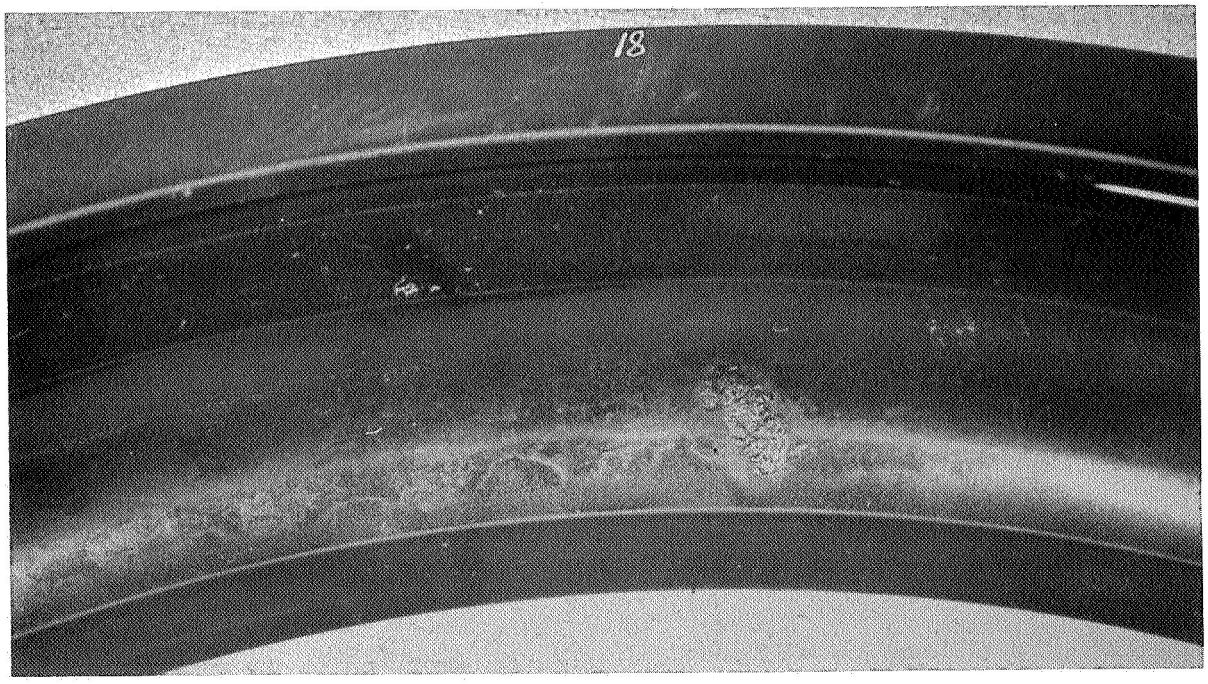


FIGURE 87. BEARING S/N 81. FATIGUE FAILURE ON OUTER RING

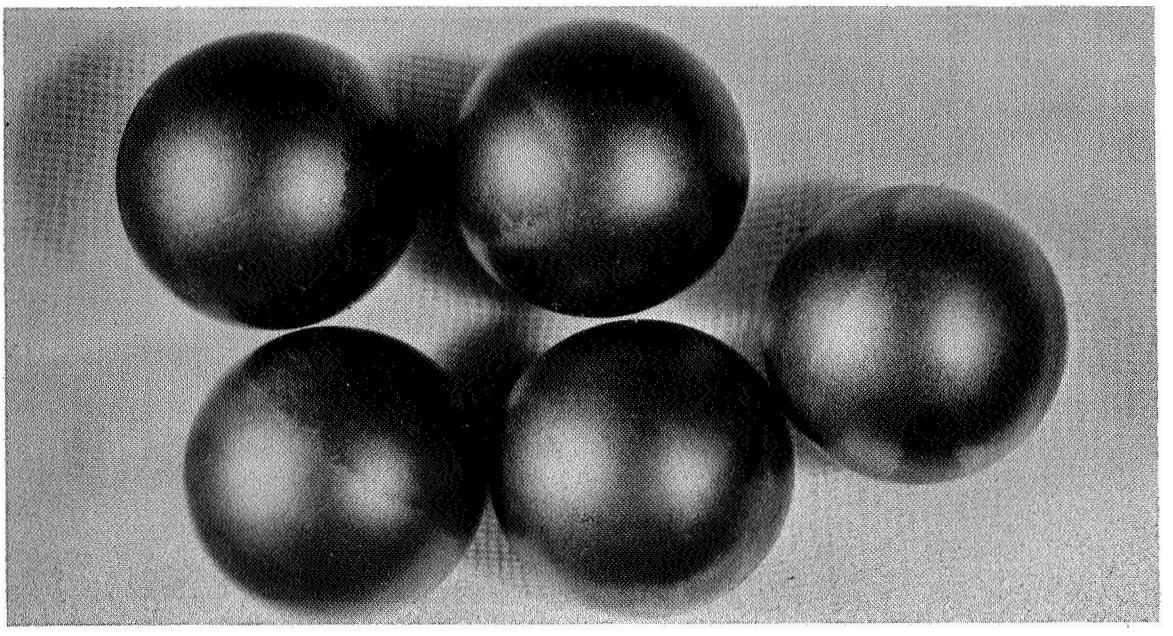


FIGURE 88. BEARING S/N 81. BALLS

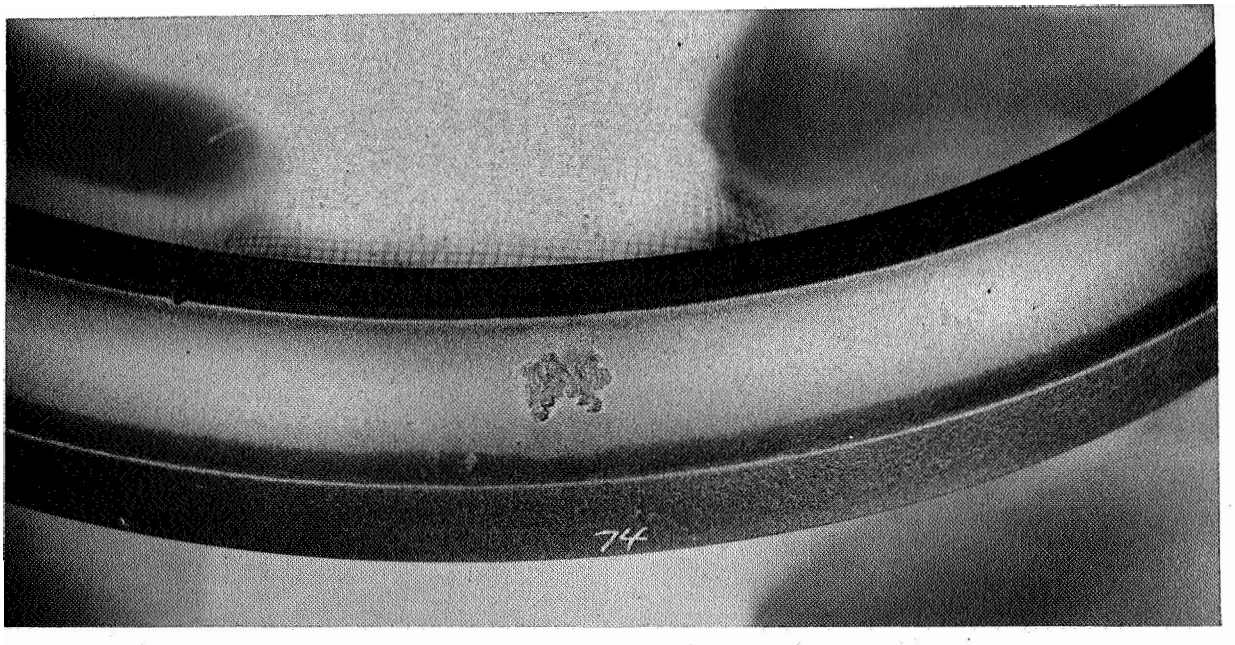


FIGURE 89. BEARING S/N 74. .FATIGUE FAILURE ON INNER RING.

TEST CONDITIONS:

LOAD :	5800 LB.
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	436 HOURS

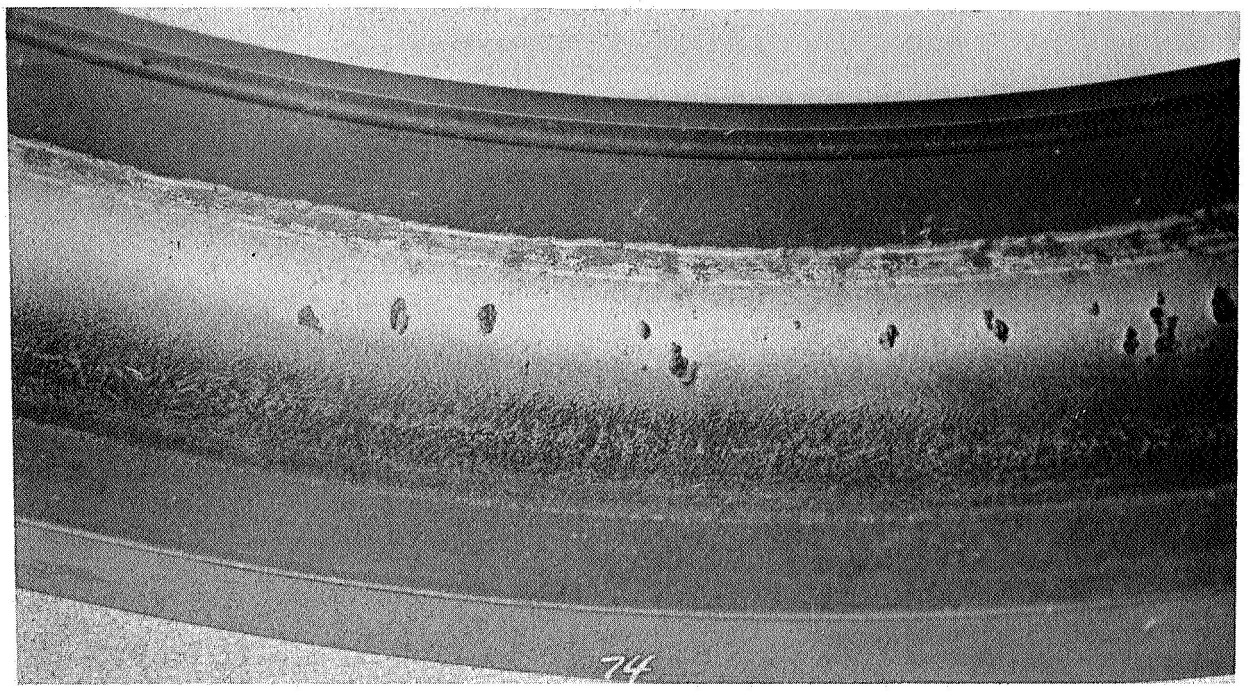


FIGURE 90. BEARING S/N 74. FATIGUE FAILURE ON OUTER RACE. NOTE EVIDENCE OF CHEMICAL CORROSION.

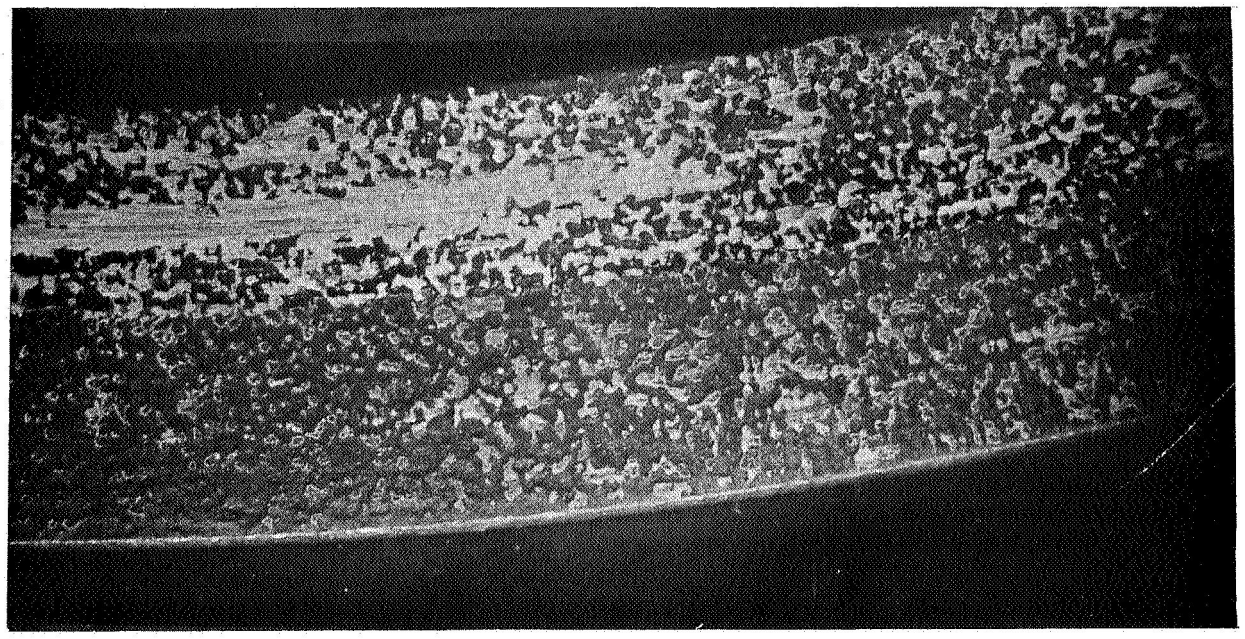


FIGURE 91. BEARING S/N 74. 10X MAGNIFICATION OF INNER RACE SHOULDER SHOWING EXTENT OF CHEMICAL CORROSION.

FIGURE 92. BEARING S/N 93. DAMAGED INNER RING

TEST CONDITIONS:

LOAD:	4365 LB.
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	159 HOURS

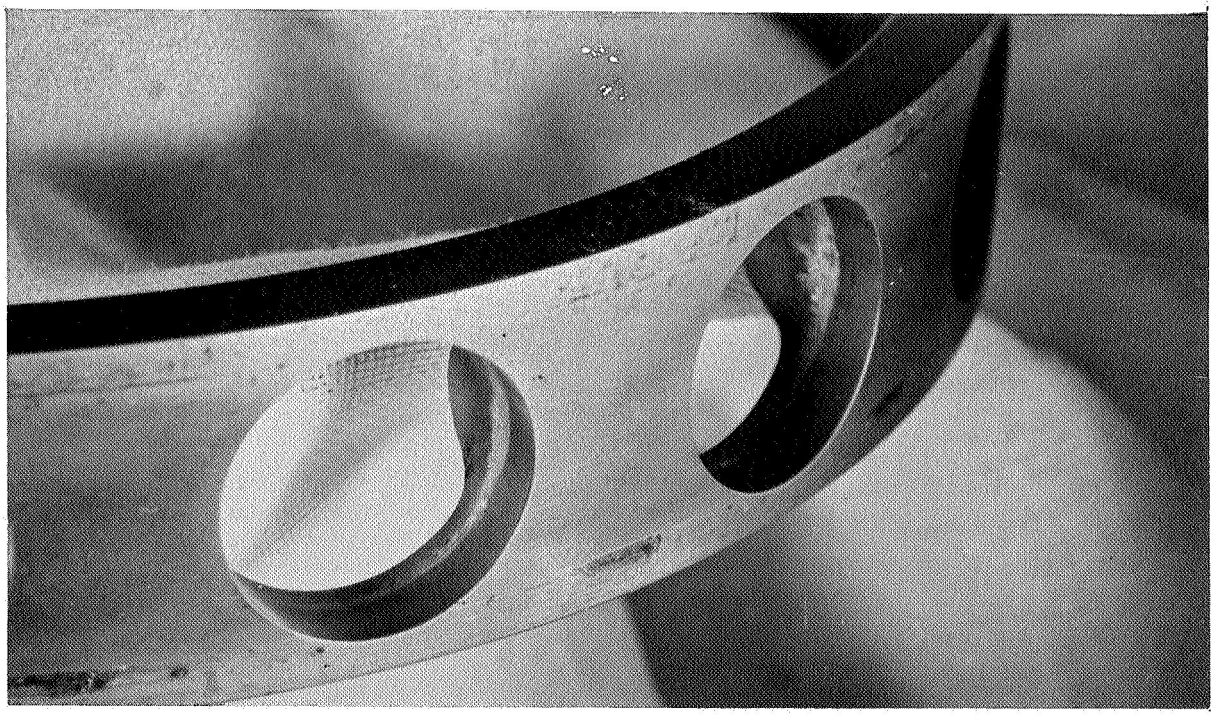


FIGURE 93. RETAINER, BEARING S/N 93

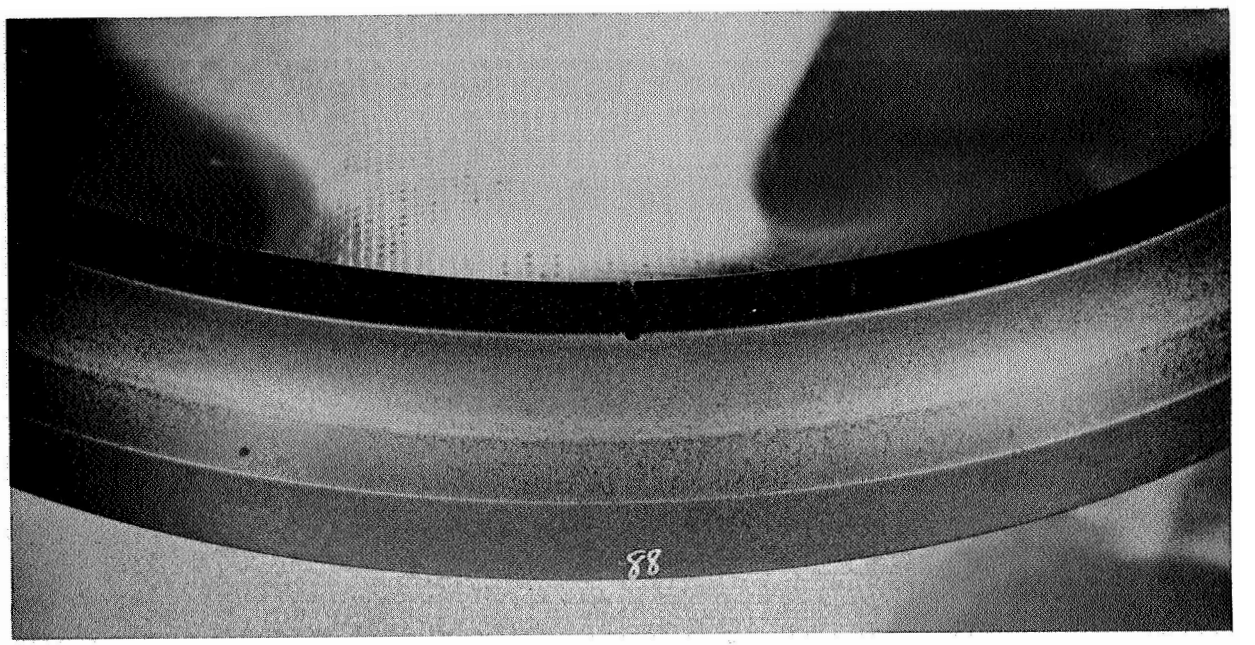


FIGURE 94. BEARING S/N 88. INNER RING.

TEST CONDITIONS:

LOAD:	4365 LB, AXIAL
SPEED:	12,000 RPM
TEMPERATURE;	600F
LUBRICANT:	PR-143

TEST TERMINATED AT 500 HOURS.
NO FAILURE.

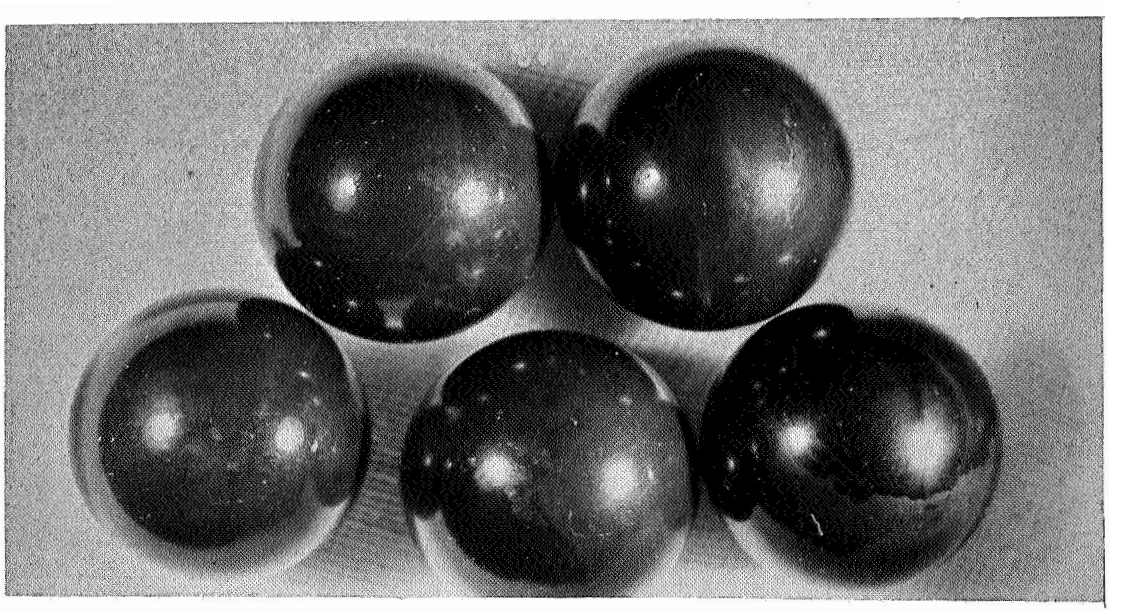


FIGURE 95. BEARING S/N 88. BALLS

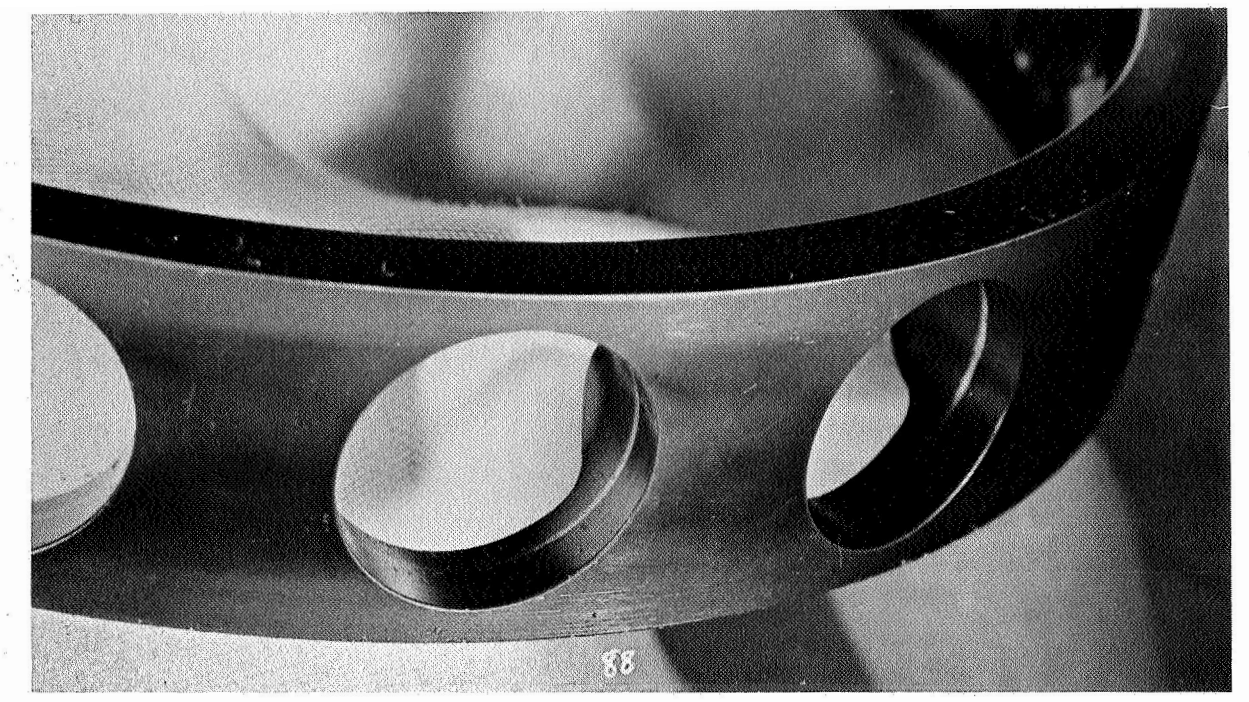


FIGURE 96. BEARING S/N 88. CONDITION OF RETAINER AFTER 500 HOURS.

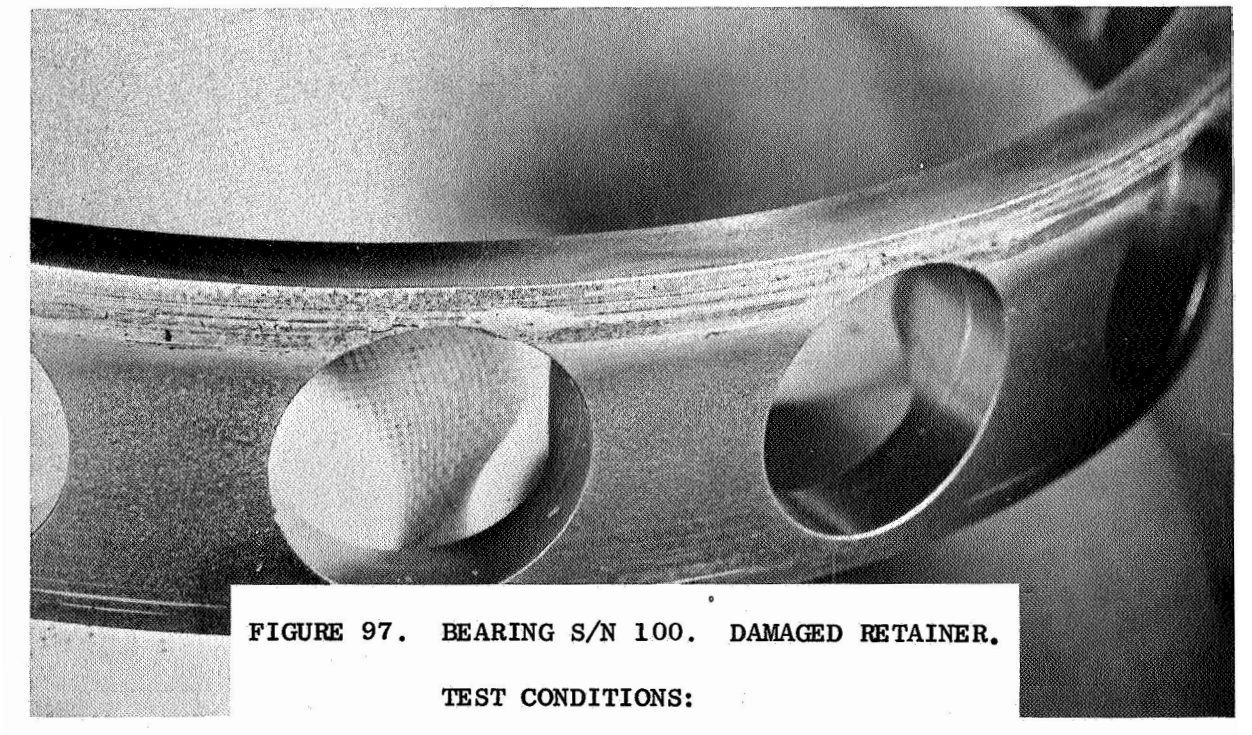


FIGURE 97. BEARING S/N 100. DAMAGED RETAINER.

TEST CONDITIONS:

LOAD:	4365 LB.
SPEED:	12,000 RPM
TEMPERATURE:	600F
LUBRICANT:	PR-143
TIME TO FAILURE:	.01 HOURS

While the expected corrosion problems did not materialize in the magnitude initially feared, there was evidence that the PR-E43 was chemically attacking the bearing material. This attack is associated with long time tests such as bearings S/N 73 and 74, where chemical attack of all of the bearing surfaces was noticeable. In the case of these two bearings, fatigue spalling, wear and chemical attack is noted. The Monel cages, however, appear to resist quite well the corrosive effects of the PR-143.

It should be pointed out, however, that the overall corrosiveness of the PR-143 at 600F was less than had been expected based on previous data.[3] The tester components which include Ni-plated steel, Monel and 300 series SS showed essentially no deterioration after exposures in excess of 1,000 hours to the fluid. This can, of course, be attributed to a large extent to the excellent inertion system of the testers which consistently maintained the O_2 level at less than 5 ppm.

One additional difficulty with the PR-143 was the evidence of poor cage lubrication. Bearing S/N 100 seized after one minute of operation due to cage pick-up on the outer land riding surface. This was the only instance of this event in all of the testing and while therefore unique, does perhaps tend to demonstrate the lack of boundary lubrication of the PR-143.

The last bearing tested (S/N 99) is perhaps typical of the type of operation which can be expected with the DuPont Fluid. This bearing showed evidence of chemical attack, ball wear and fatigue. Ball wear was on the order of .0015" and an etching effect was plainly visible in the running track,

6.2.4 Test Series IV

The conditions for this test series were:

Bearing Material:	Rings:	WB-49
	Balls:	M-1
	Cages;	M-1
Lubricant:	Mobil XRM-177F	
Operating Conditions:		
	Temp:	Outer Ring 585F ± 5F
		Inner Ring 600F - 610F
		Oil In - 545F-565F
		Oil Out - 585F-600F
	Speed:	12,000 rpm
	Load:	5800 lbs. axial
	Atmosphere:	N_2 <5 ppm O_2

Results of the bearing tests are presented in Table 15 and graphically illustrated in Weibull distribution in Figure 98.

A trend of low fatigue life was established with this material early in the test program. The reason for this appears to be related to the microstructure of the material as will be discussed later.

The failures encountered were with one exception, normal sub-surface initiated fatigue failures. Figures 99 through 108 are photographs of typical WB-49 bearing components failed in fatigue. The only deviation from these failures was in bearing S/N 272, where a circumferential outer ring fracture was observed. Generally, the balls made out of M-1 material appear to be relatively undamaged despite the rather severe fatigue failures observed on other components. This is illustrated in the bottom of Figure While there have been a number of ball fatigue failures, it is difficult to establish whether these failures are unique or are caused by a prior failure of the inner or outer ring. As may be noted in Table 15 the ball

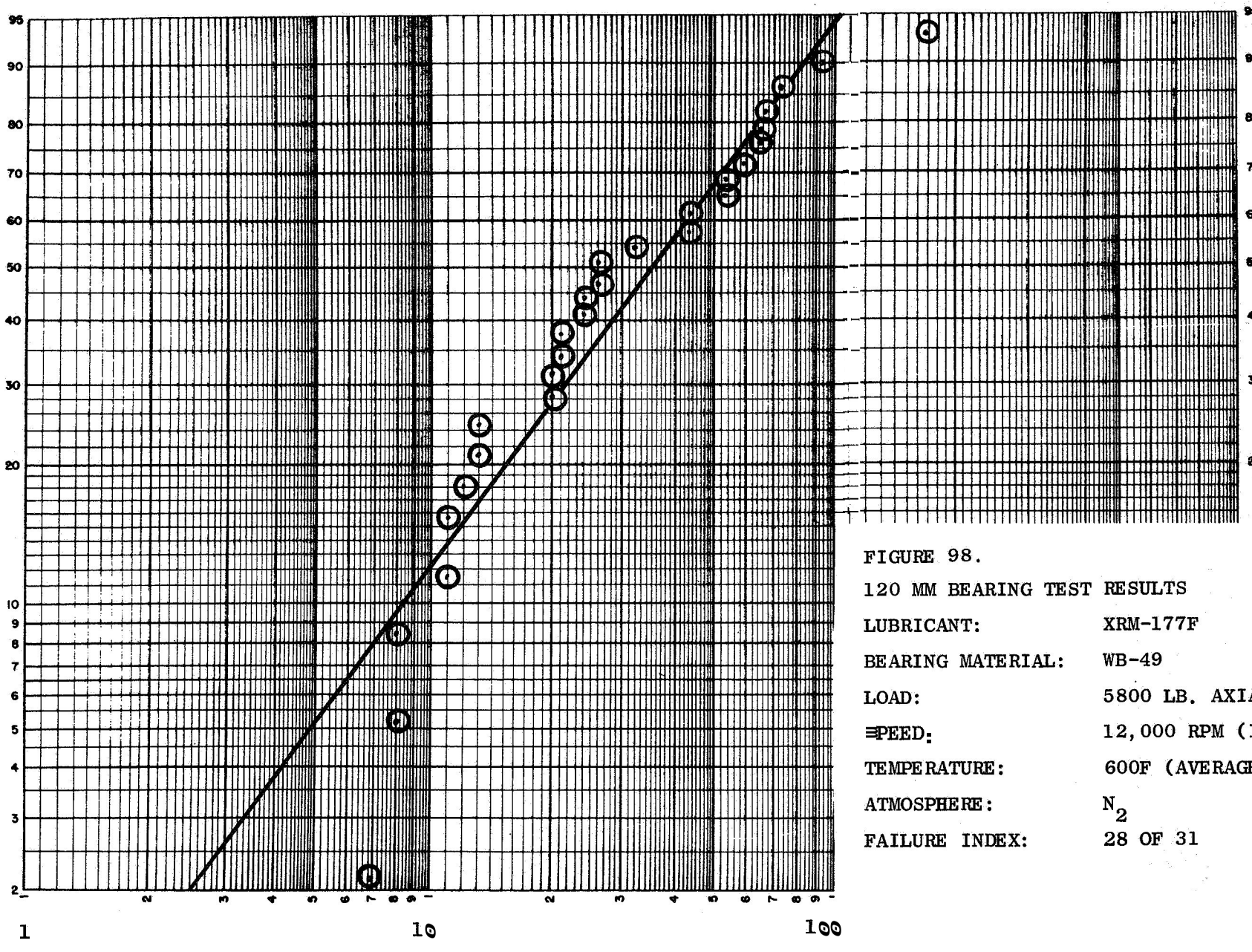


FIGURE 98.

120 MM BEARING TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	WB-49
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	600F (AVERAGE METAL)
ATMOSPHERE:	N_2
FAILURE INDEX:	28 OF 31

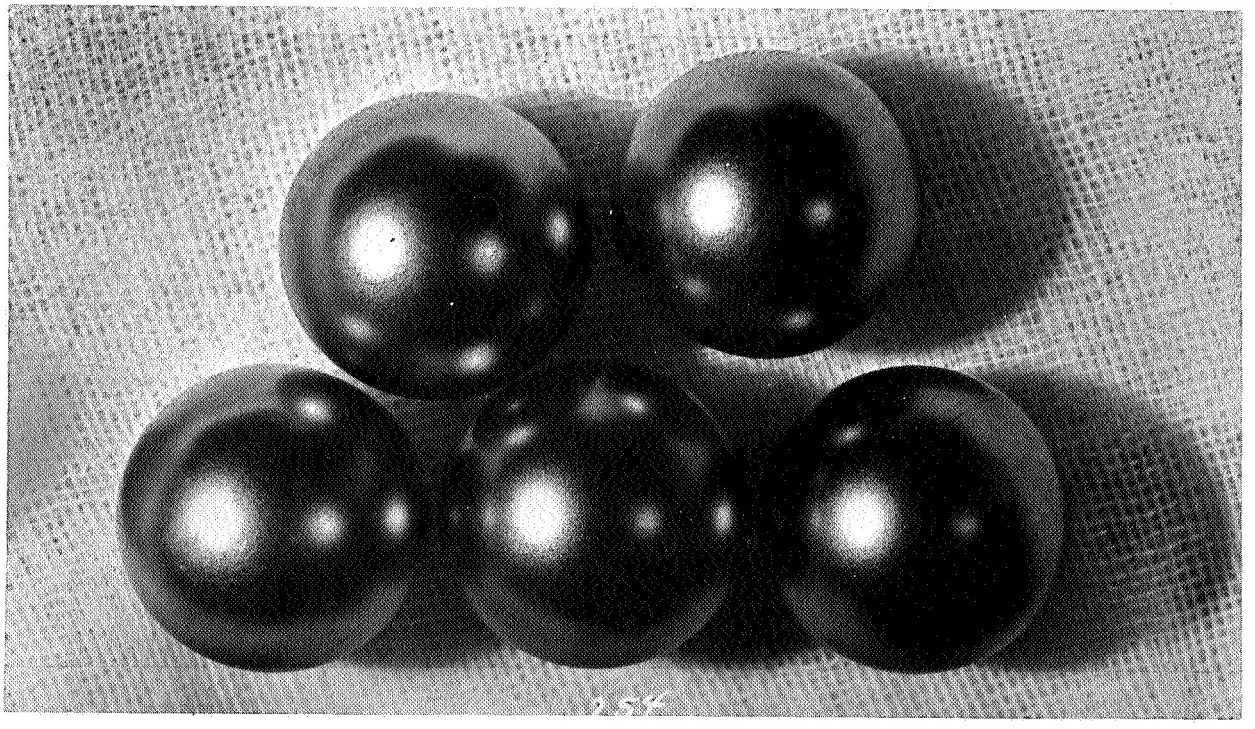
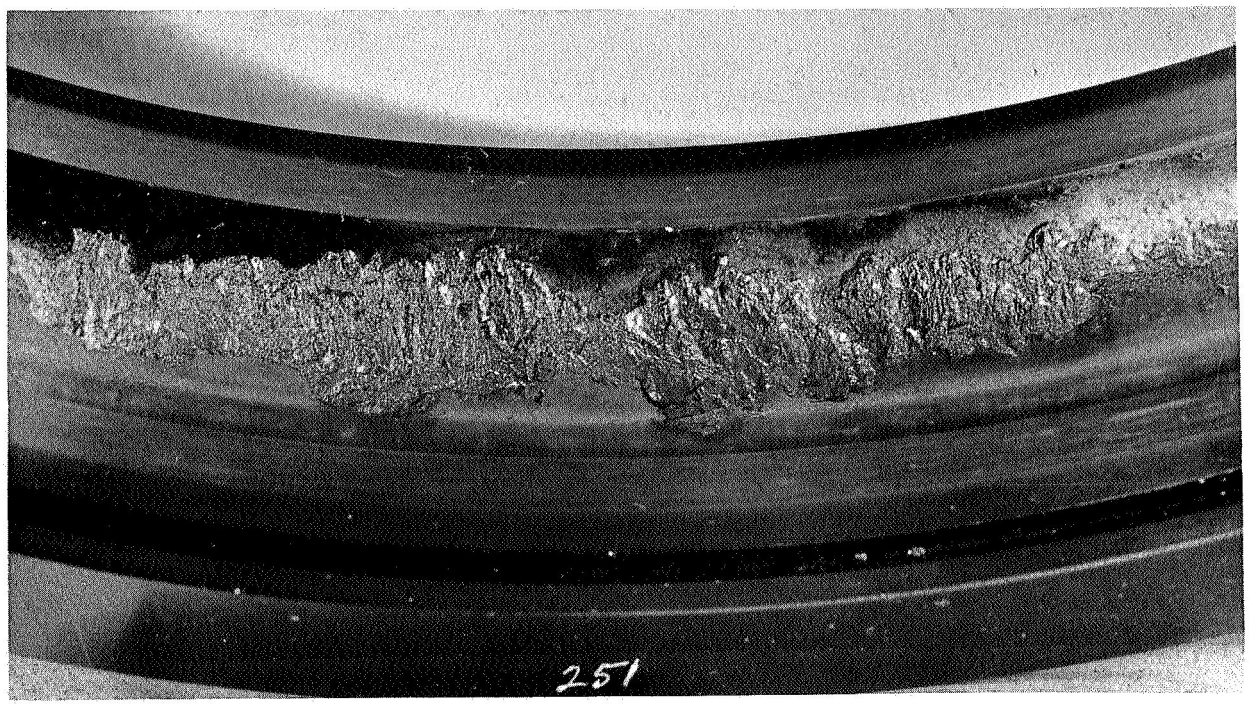


FIGURE 99. TOP: INNER RING FATIGUE FAILURE, S/N 254
BOTTOM: BALLS FROM S/N 254

BEARING MATERIAL:	WB-49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F (METAL)
TIME TO FAILURE:	26 HOURS



FIGURF: 100. OUTER RING FAILURE, S/N 251

BEARING MATERIAL:	WB-49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 **RPM**
TEMPERATURF::	600F (METAL)
TIME TO FAILURE:	8 HOURS

FIGURE 101. INNER RING FATIGUE FAILURE, S/N 256

BEARING MATERIAL:	WB–49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F (METAL)
TIME TO FAILURE:	26 HOURS

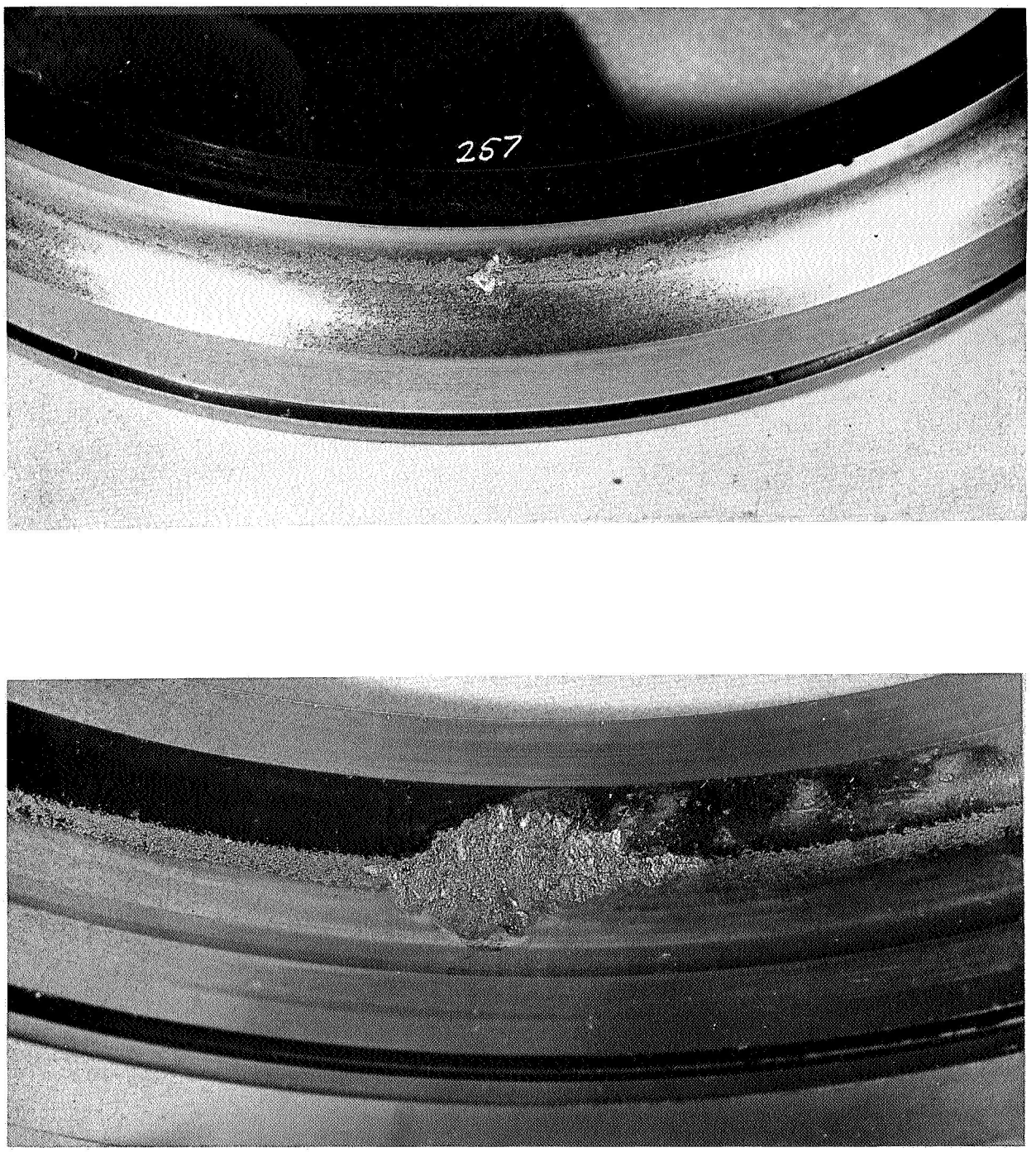


FIGURE 102. INNER AND OUTER RING FATIGUE FAILURES, S/N 257

BEARING MATERIAL:	WB-49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F (METAL)
TIME TO FAILURE:	24 HOURS

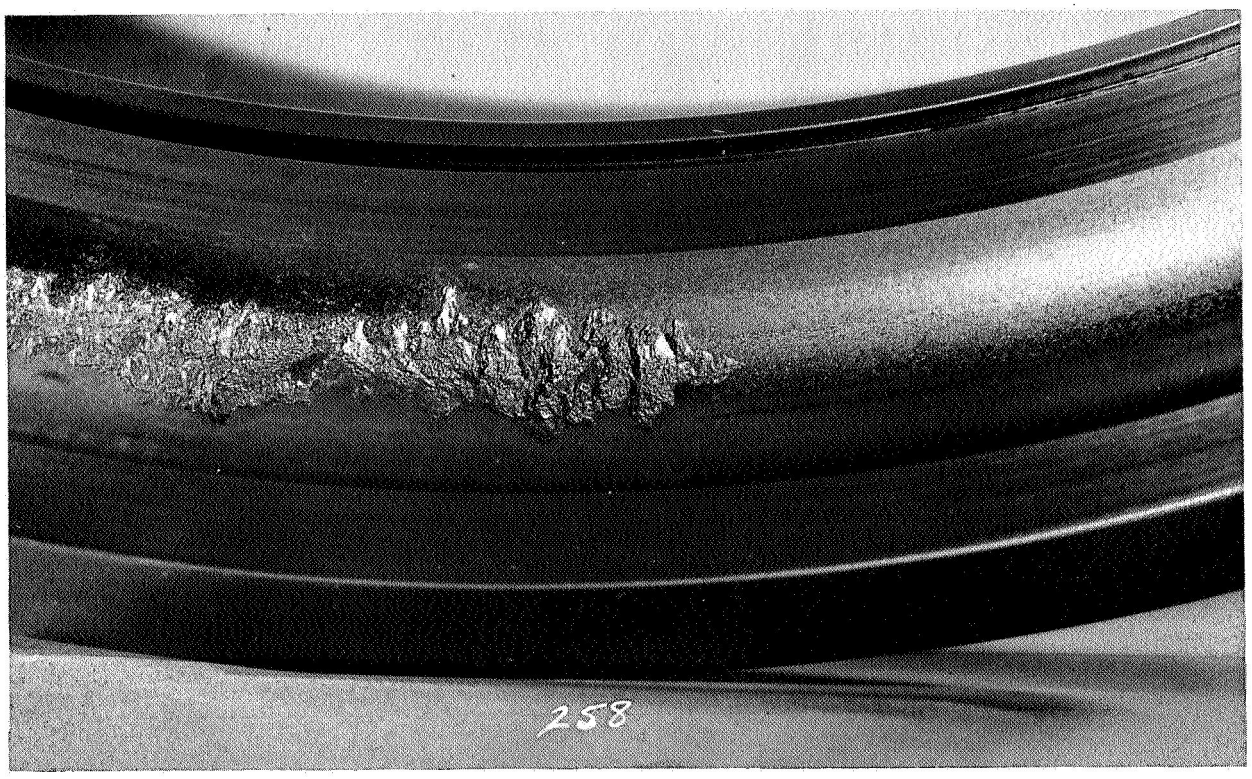


FIGURE 103. OUTER RING FATIGUE FAILURE, S/N 258

BEARING MATERIAL:	WB–49
	M-1 BALLS
LUBRICANT:	XRM–177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F (METAL)
TIME TO FAILURE:	24 HOURS

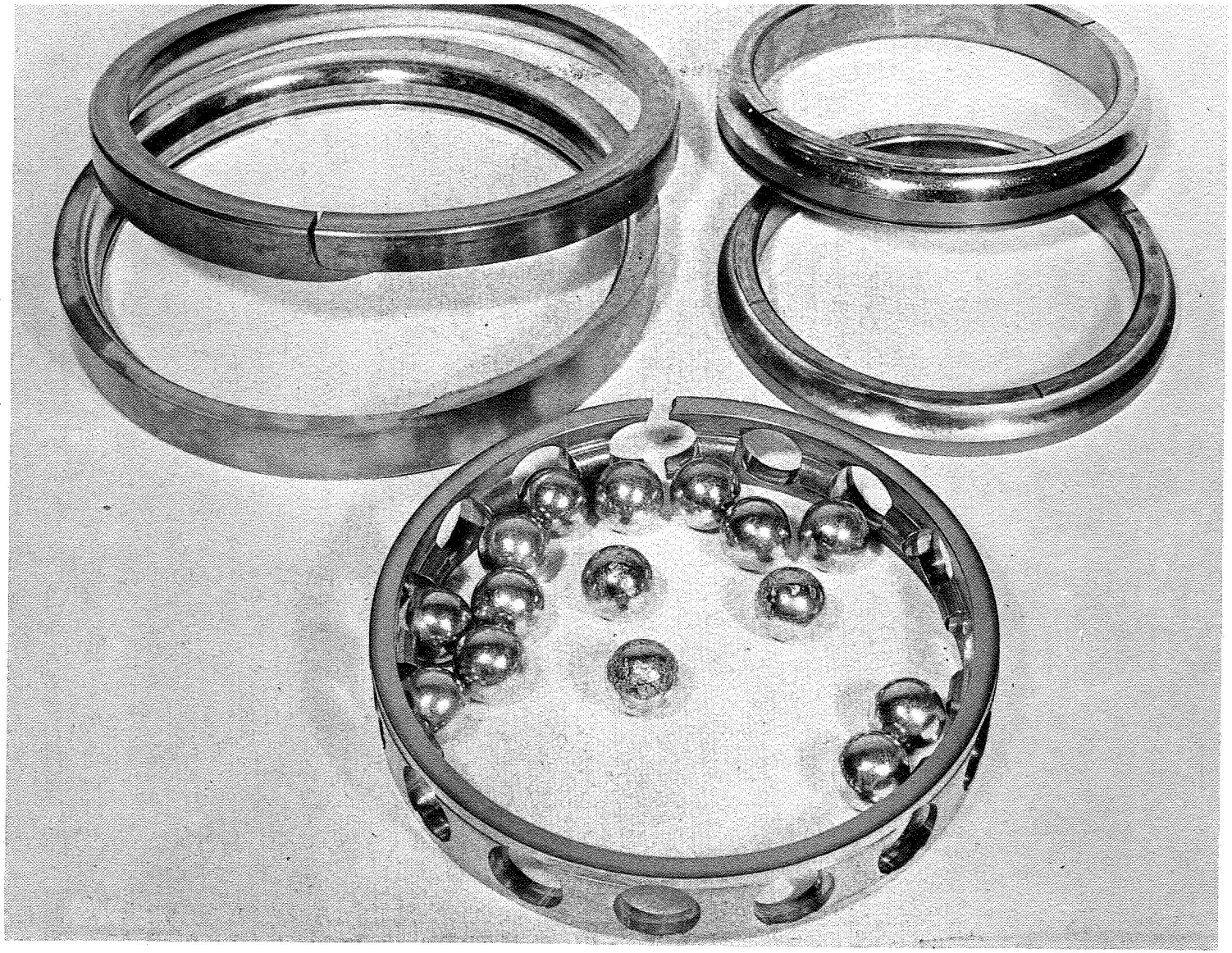


FIGURE 104. BEARING DN 272

MATERIAL:	WB-49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F (N_2 ATMOSPHERE)
TEMPERATURE:	600F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TIME TO FAILURE:	13 HOURS

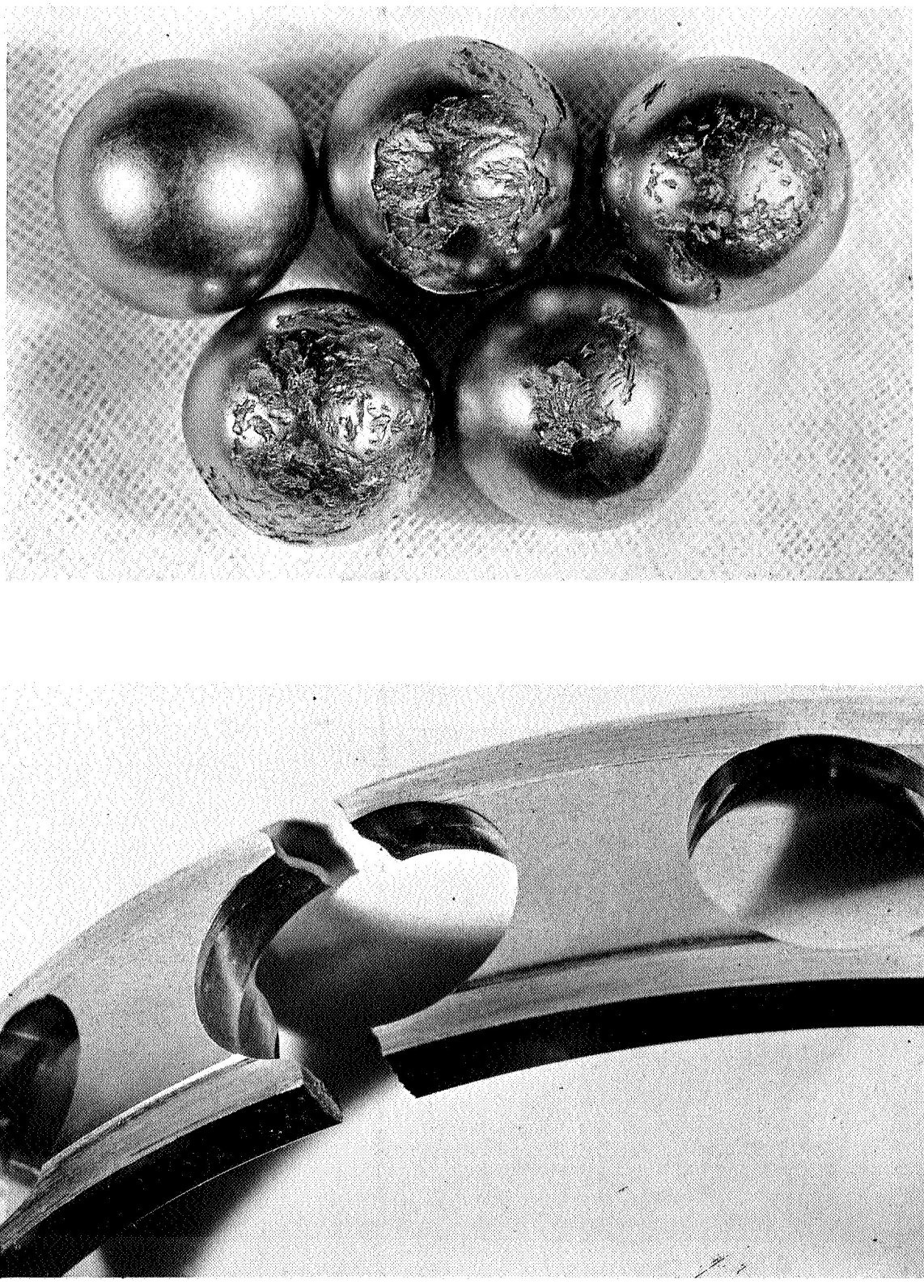


FIGURE 105. BEARING S/N **272.**

BALL FATIGUE AND CAGE FRACTURE AFTER **13 HOURS.**
NOTE EXTENSIVE DEFORMATION IN **CAGE** BALL POCKET.

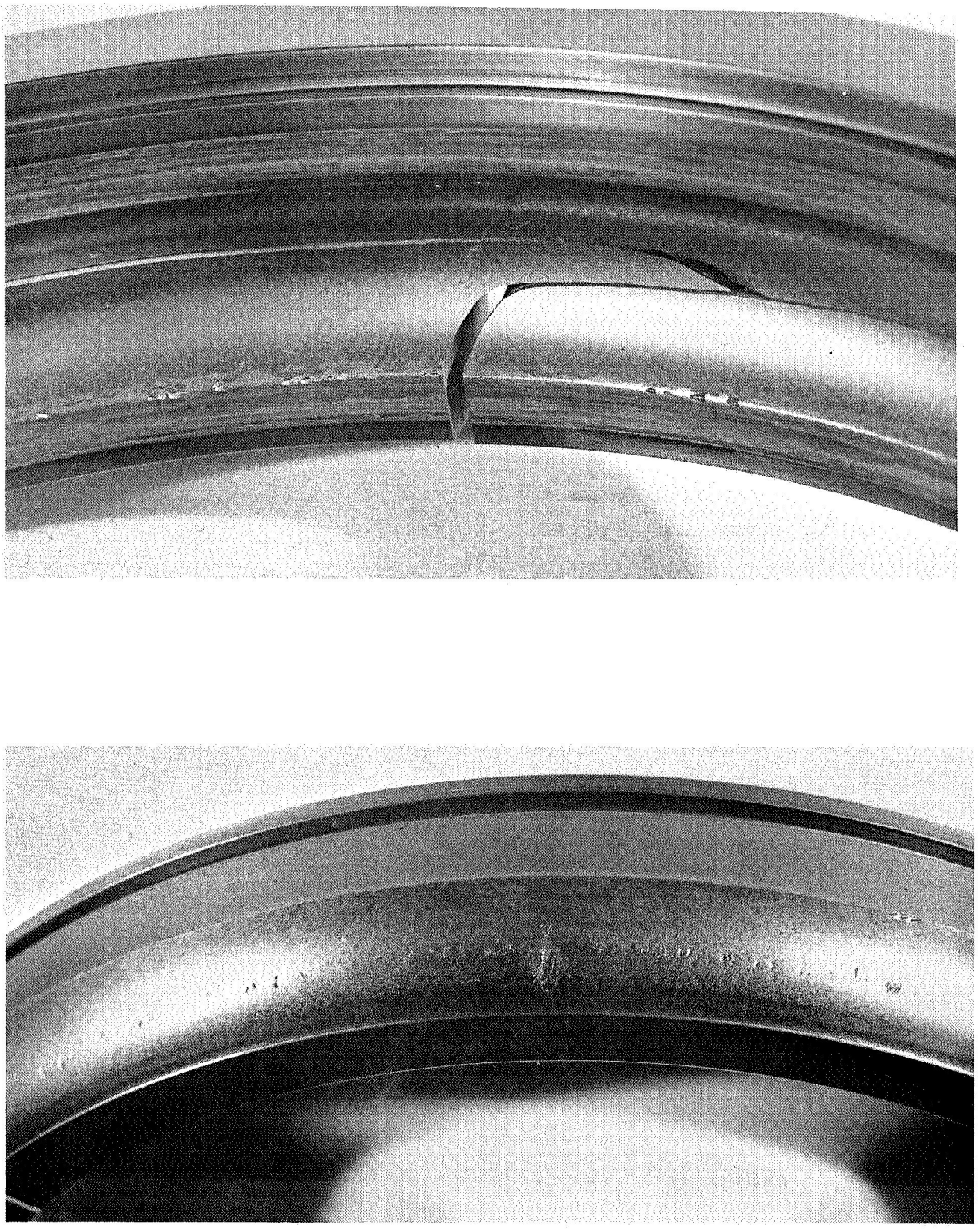


FIGURE 106. BEARING S/N 272.

CLOSE-UP OF FRACTURE ON OUTER RING AND FATIGUE FAILURES ON INNER RING.

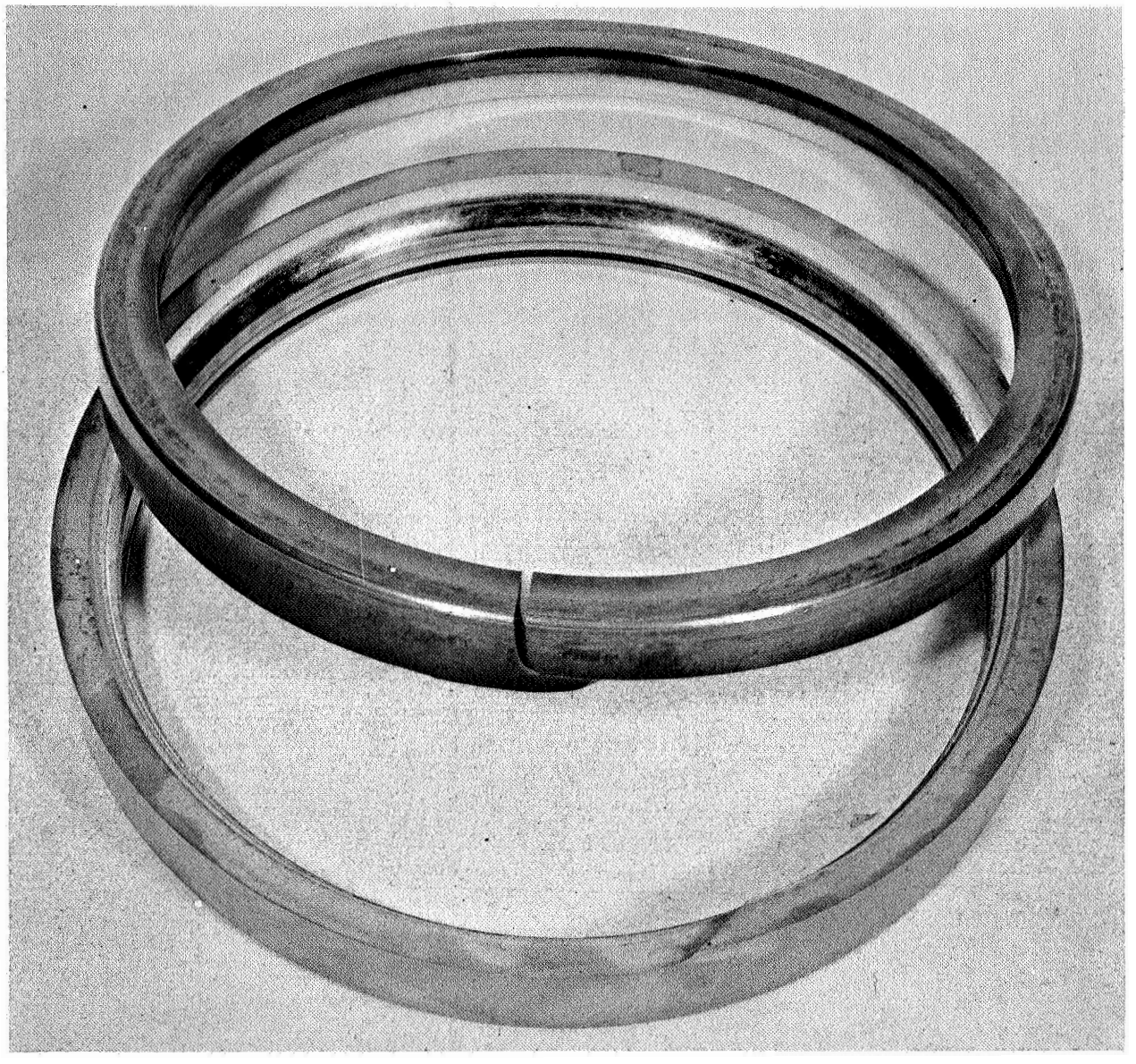
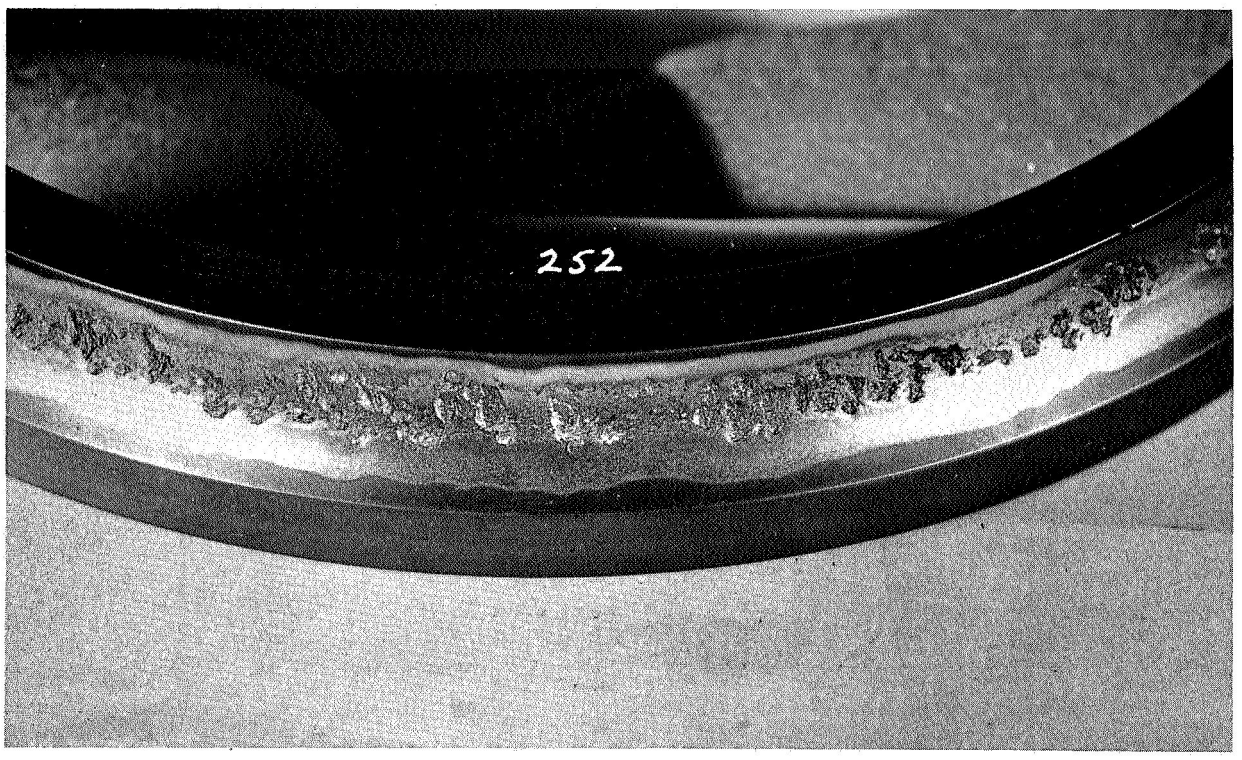


FIGURE 107. BEARING S/N 272.

CLOSE-UP OF CIRCUMFERENTIAL OUTER RING FRACTURE AFTER 13 HOURS.



FIGURF 108. . INNER RING FATIGUE FAILURF, S/N 252

BEARING MATERIAL:	WB-49 RINGS M-1 BALLS
LUBRICANT:	XRM-177F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
TEMPERATURE:	600F (METAL)
TIME TO FAILURE:	8 HOURS

failures have always occurred in conjunction with another component failure or in a bearing having received excessive foreign object damage,

Of particular interest in this test series were the M-1 steel cages which were used instead of the S-Monel in order to evaluate their behavior at 600F. While most of the cages performed satisfactorily, it was found that eight of the cages fractured similar to the one shown in Figure 105. This is a far greater incidence of cage fractures than had been observed in the previous test where S-Monel was used as the structural material for these parts. It was also observed that the M-1 cages had a greater amount of ball pocket wear than had been observed on the S-Monel cages, The data would tend to indicate that at least for the higher temperature applications, S-Monel may be superior to M-1 steel for use as a cage material.

The overall low fatigue life exhibited by the WB-49 bearings had been predicted by RC rig testing.

6.2.5 Test Series V

The following conditions were observed in this test series:

Bearing Material:	Rings:	CVM M-50
	Balls:	CVM M-50
	Retainer:	S-Monel
Lubricant:	Mobil XRM-177F	
Operating Conditions:		
	Temp:	Outer Ring 490F - 500F
		Inner Ring 505F - 515F
		Oil In - ~482F
		Oil Out - ~518F
	Speed:	12,000 rpm
	Load:	5800 lbs. axial
	Atmosphere:	N_2 <5 ppm O_2

Test results are presented in Table 16 and a Weibull plot of the data is shown in Figure 109.

The Weibull plot is based on a failure index of 11 out of 26. Several of the extremely short time failures have been eliminated in the analysis as there was sufficient evidence in each case of foreign object damage. The foreign object damage and the small amount of ball wear which was observed were traoed to debris from pump damage due to the PR-143,

In this test series, the cut-off point was raised to 750 hours rather than the 500-hour limit observed in the previous tests, as it was reasoned that the increased viscosity at the lower testing temperature would provide a better elastohydrodynamic lubrication.

Table 16

Results of 120 mm Bearing Tests with XRM-177F Lubricant, CVM M-50 Bearings, at 500F in N_2

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
V - 1	107	5800 (1)	48	OK	OK	OK (.0005" Wear)
V - 1a	108	5800	48	OK	OK	OK (.0005" Wear)
V - 2	105	5800	256	OK	OK	OK (.0002' Wear)
V - 2a	106	5800	256	Fatigue	OK	OK (.0002" Wear)
V - 3	109	5800	226	OK	OK	OK (.0004" Wear)
V - 3a	110	5800	226	OK	OK	OK (.0004' Wear)
V - 4	119	5800	20	OK	OK	OK
V - 4a	120	5800	20	OK (FOD)	OK	Fatigue
V - 5	103	5800	536	OK	OK	OK
V - 5a	104	5800	536	OK	OK	OK
V - 6	111	5800	573	OK	OK	OK
V - 6a	112	5800	573	OK	Fatigue	OK
V - 7	121	5800	373	OK	OK	OK
V - 7a	122	5800	373	Cage Break	Wiped	Seized
V - 8	115	5800	508	OK	OK	OK
V - 8a	116	5800	508	Fatigue	OK	Fatigue
V - 9	127	5800	8	Fatigue (FOD)	OK	OK
V - 9a	128	5800	8	OK	OK	OK
V -10	103	5800	558	OK	OK	OK (.0003" Wear)
V -10a	104	5800	558	OK	OK	OK (.0002" Wear)
V -11	117	5800	661	OK	OK	OK (.0002" Wear)
V -11a	118	5800	661	Fatigue	OK	OK

Table 16, Cont'd

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
V -12	113	5800	758	OK	OK	Fatigue
V -12a	114	5800	758	OK	OK	Fatigue
V -13	115	5800	529	Fatigue	OK	Multiple Fatigue
V -13a	128	5800	9	OK	OK	OK
V -14	117	5800	663	OK	OK	OK
V -14a	128	5800	12	Fatigue (FOD)	OK	Multiple Fatigue
V -15	121	5800	677	OK	Multiple Fatigue	Multiple Fatigue
V -15a	111	5800	877	Fatigue	OK	OK
V -16	123	5800	751	OK	OK	OK (.0002" Wear)
V -16a	124	5800	751	OK	Fatigue	OK (.0003' Wear)
V -17	125	5800	729	OK	OK	Fatigue
V -17a	126	5800	729		Plugged Jet	

(1) 5800 lb. axial load = 321,000 psi max. hertz on inner.

☐ Indicates Time to Fatigue Failure

FOD = Foreign Object Damage

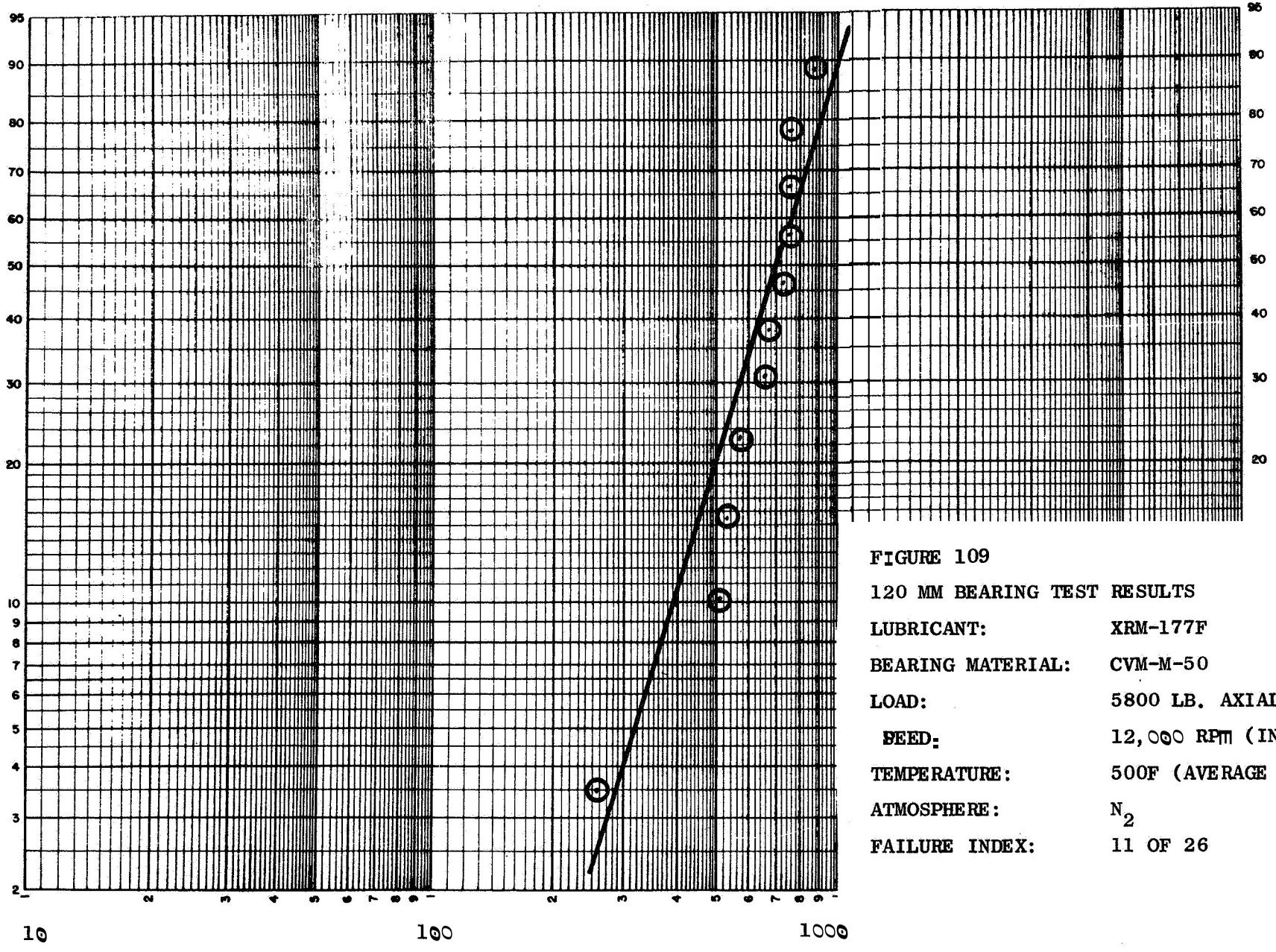


FIGURE 109

120 MM BEARING TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	CVM-M-50
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	500F (AVERAGE METAL)
ATMOSPHERE:	N_2
FAILURE INDEX:	11 OF 26

6.2.6 Test Series VI

Test conditions for this series were as follows:

Bearing Material:	Rings:	CVM-M-50
	Balls:	CVM-M-50
	Retainer:	S-Monel
Lubricant,:	Mobil XRM-177F	
Operating Conditions:		
	Temp:	Outer Ring 395F - 410F
		Inner Ring 405F - 415F
		Oil In - 370F
		Oil Out - 400F
	Speed:	12,000 rpm
	Load:	5800 lb. axial
	Atmosphere:	N_2 5 ppm O_2

The results of the tests are presented in Table 17 and shown in Weibull distribution in Figure 110. Three tests (#2, 2a, 6) were eliminated from the analysis. The first two because the test was aborted due to a plugged lube jet and the third one because the bearing had superficial foreign object damage because of the plugged jet,

The photographs in Figures 111 through 113 illustrate the typical condition of the M-50 bearings after prolonged operation at 400F in nitrogen inerted XRM-177F. As can be seen from the photographs, the bearing components are in excellent condition, Of particular interest is the fact that several of the bearings in this test series including the one shown in the referenced photographs had been given a black oxide coating (Duliting) in order to more sharply define the ball track and thus the operating oontact angle. The oxide coating apparently had no effect on the bearing life, although it did serve to accurately delineate the ball path as is illustrated in Figure 113,

Again, as in Test Series V, cut-off was at 750 hours rather than 500 hours.

Table 17

Results of 120 mm Bearing Tests with XRM-177F Lubricant, CVM M-50 Bearings, at 400F in N_2

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
VI - 1	131	5800	278	Fatigue		Fatigue
VI - 1a	132	5800	278	OK	OK	OK
VI - 2	145	5800	.1	OK	OK	OK
VI - 2a	146	5800	.1	Plugged Jet	Pickup	
VI - 3	133	5800	163	Fatigue	OK	Fatigue
VI - 3a	134	5800	163	OK	OK	OK
VI - 4	129	5800	755	OK	OK	OK
VI - 4a	130	5800	755	OK	OK	OK
VI - 5	149	5800	31	OK	OK	OK
VI - 5a	134	5800	194	Fatigue	FOD	Fatigue
VI - 6	145	5800	1	Fatigue	FOD	Fatigue
VI - 6a	132	5800	280	OK	OK	OK (.0003" Wear)
VI - 7	135	5800	778	OK	OK	OK
VI - 7a	136	5800	778	OK	OK	OK
VI - 8	137	5800	752	OK	OK	OK
VI - 8a	138	5800	752	OK	OK	OK
VI - 9	141	5800	683	Fatigue	OK	OK
VI - 9a	142	5800	683	FOD	OK	OK
VI -10	139	5800	768	OK	OK	OK
VI -10a	140	5800	768	OK	OK	OK
VI -11	147	5800	775	OK	atig	OK
VI -11a	148	5800	775	OK	OK	OK

Table 17, Contd.

Test No.	Bearing S/N	Load (lb.)	Time (Hr.)	Inner Ring	Outer Ring	Ball
VI -12	143	5800	788	Fatigue	OK	OK
VI -12a	144	5800	788	OK	OK	OK
VI -13	132	5800	828	OK	Fatigue	OK (.0003" Wear)
VI -13a	149	5800	744	FOD	OK	Fatigue
VI -14	170	5800	554	Fatigue	FOD	Fatigue
VI -14a	171	5800	554	OK	Fatigue	OK
VI -15	172	5800	767	OK	OK	OK
VI -15a	173	5800	767	OK	OK	OK

(1) 5800 lb. axial load = 321,000 psi max. hertz on inner

(2) FOD = Foreign Object Damage

□ Indicates Time to Fatigue Failure

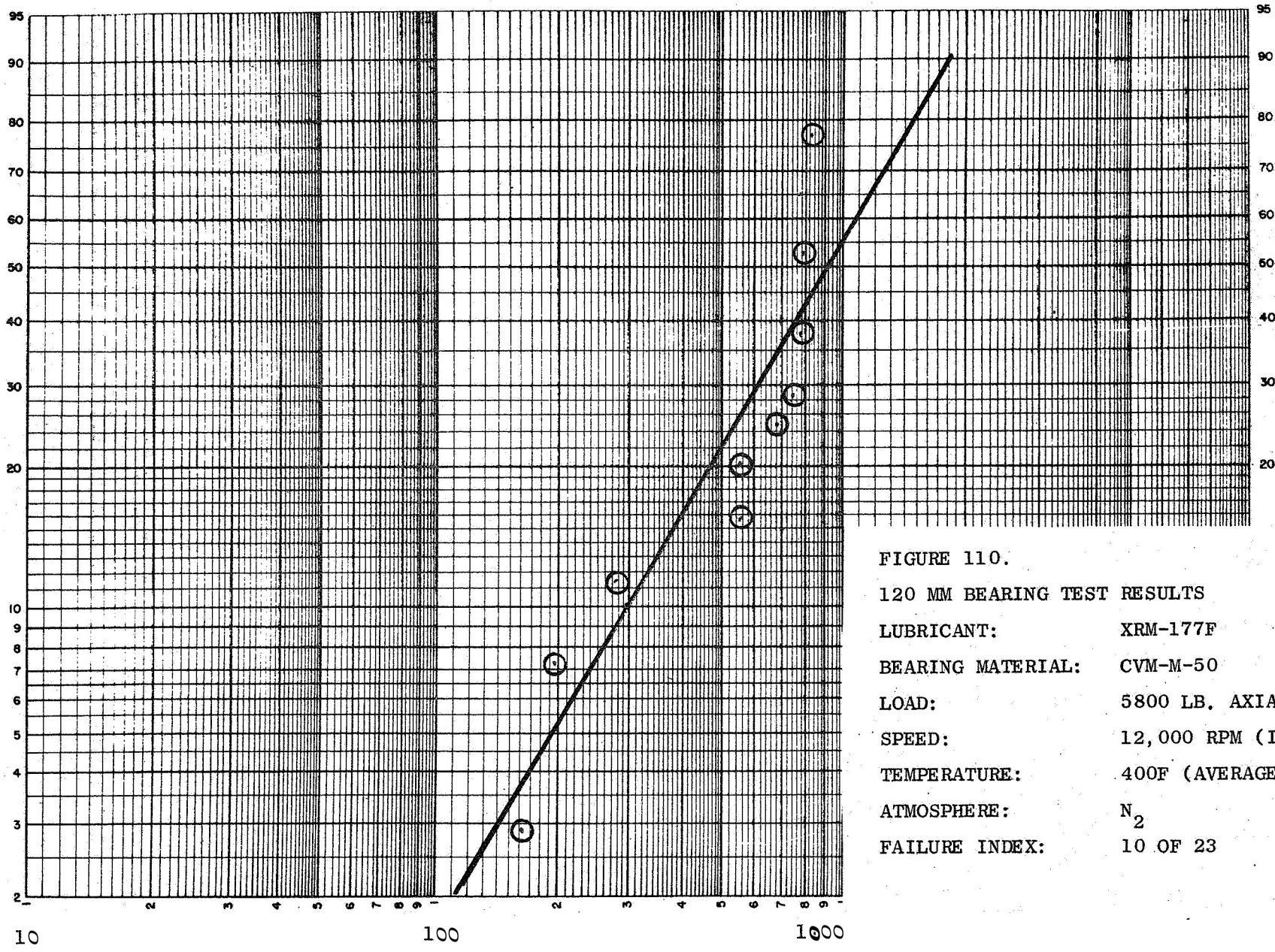


FIGURE 110.

120 MM BEARING TEST RESULTS

LUBRICANT:	XRM-177F
BEARING MATERIAL:	CVM-M-50
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM (INNER)
TEMPERATURE:	400F (AVERAGE METAL)
ATMOSPHERE:	N_2
FAILURE INDEX:	10 OF 23

FIGURE 111. BEARING S/N 172

LUBRICANT:	XRM-177F (N_2 ATMOSPHERE)
TEST TEMPERATURE:	400F
LOAD:	5800 LB. AXIAL
SPEED:	12,000 RPM
MATERIAL:	M-50 BALLS AND RINGS
	S-MONEL CAGE

TEST SUSPENDED AT 767 HOURS

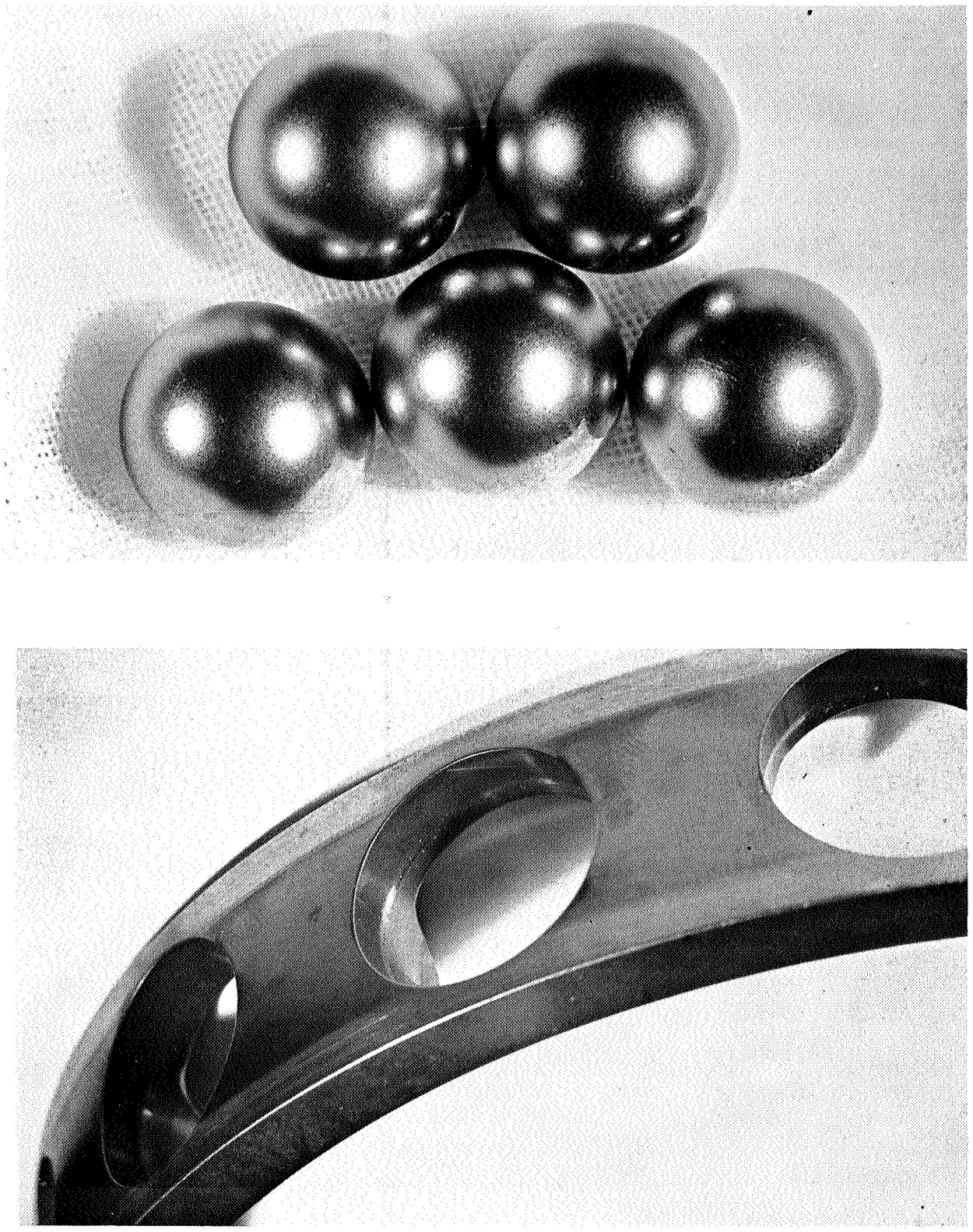


FIGURE 112. BEARING S/N 172.

APPEARANCE OF BALLS AND RETAINER
AFTER 767 HOURS AT 400F WITH XRM-177F.

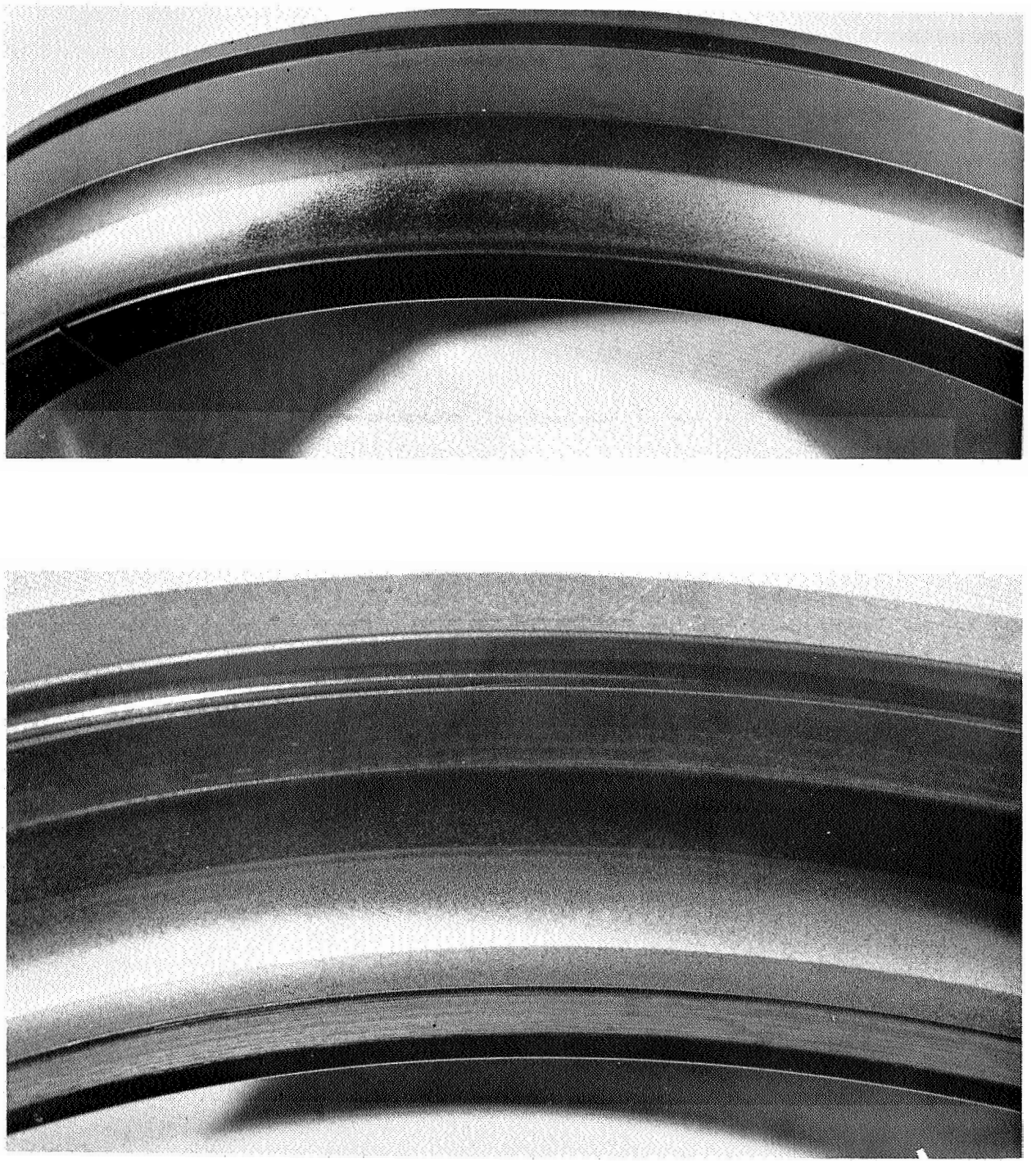


FIGURE 113. BEARING S/N 172.

APPEARANCE OF INNER RING (TOP) AND OUTER RING (BOTTOM) BALL PATH AFTER 767 HOURS AT 400F WITH XRM-177F.

7.0 Metallurgical Investigation

Metallurgical analysis of the bearing failures did not reveal any unusual or unexpected conditions with the exception of the WB-49. The failures obtained at 600F with the three test fluids were generally typical sub-surface initiated spalling fatigue, very similar to the failures obtained under more moderate temperature conditions. The photographs in Figures 114 through 124 are presented as representative of the micro-structures observed.

In general, the photographs in the subject figures are self-explanatory, although there are a few specific comments which can be made relative to some of the photomicrographs. For example, the ball fatigue failure shown in Figure116 is somewhat unique in regard to the multiplicity of the cracks observed. Based on the RECAP analysis which indicates that the depth of maximum sub-surface shear stress should be approximately .010" below the surface, the indications are that the primary failure is the crack farthest below the surface with the remaining cracks being secondary and possibly being more of a brittle shattering nature, rather than fatigue.

Figure117 is a cross-section through an area of ball wear, the wear being evident by the relative unevenness of the ball surface.

Figure 118 is presented as a record of the micro-structure of the M-50 material being tested and shows this material to be the normal fine grained martensitic structure judged optimum for bearing applications. The primary purpose for exhibiting Figure 119 is to show the relative lack of surface deformation or micro-structural damage in a ball which had been shown to have a slight amount of wear. Again, note the unevenness of the surface. Despite this, however, there appears to be no micro-structural damage at or near the surface.

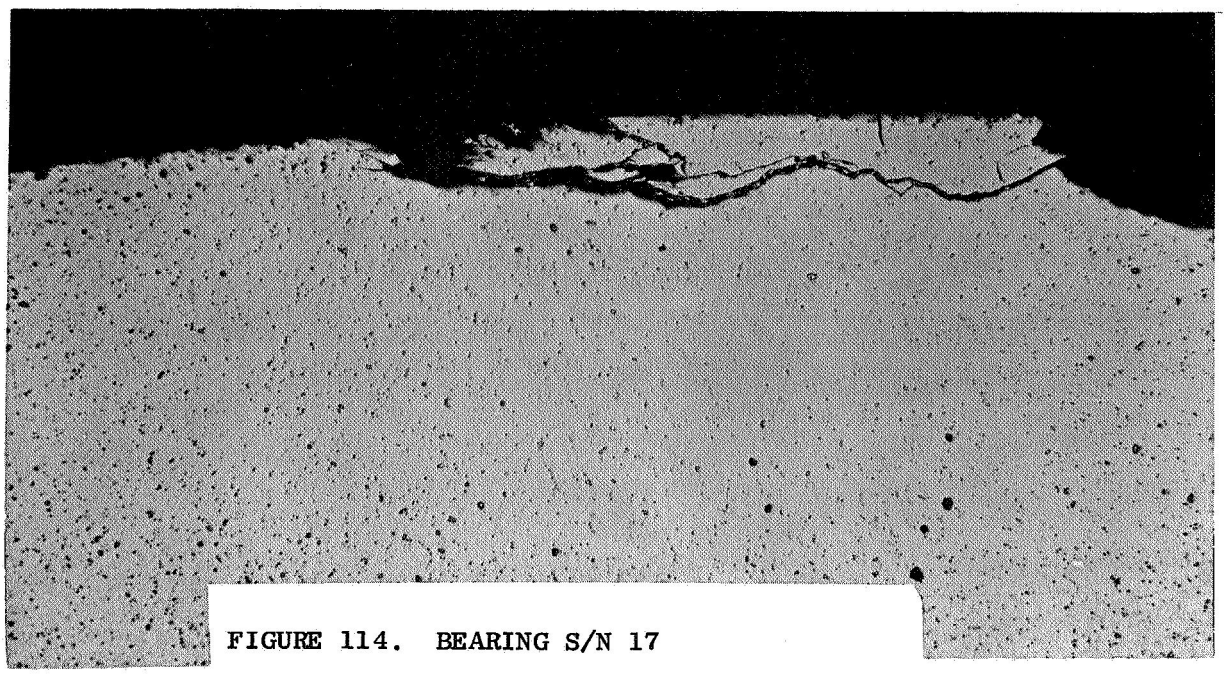


FIGURE 114. BEARING S/N 17

375 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM-177F
BALL FATIGUE FAILURE
MATERIAL: CVM-M-50
MAG: 100X UNETCHED

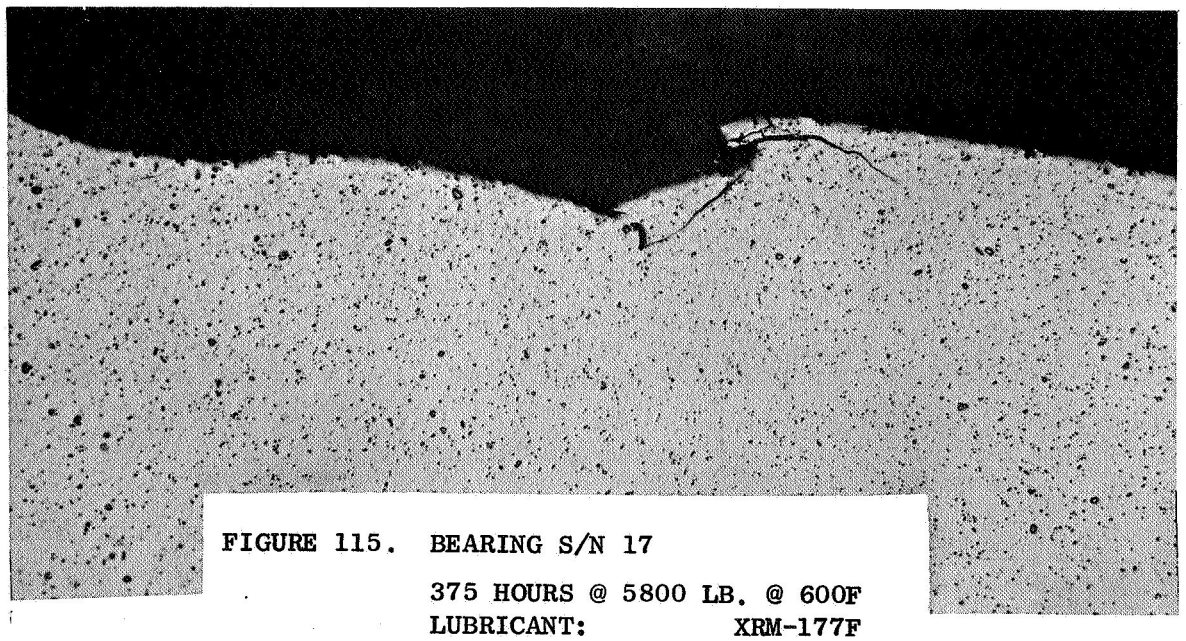


FIGURE 115. BEARING S/N 17

375 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM-177F
BALL FATIGUE FAILURE
MATERIAL: CVM-M-50
MAG: 100X UNETCHED

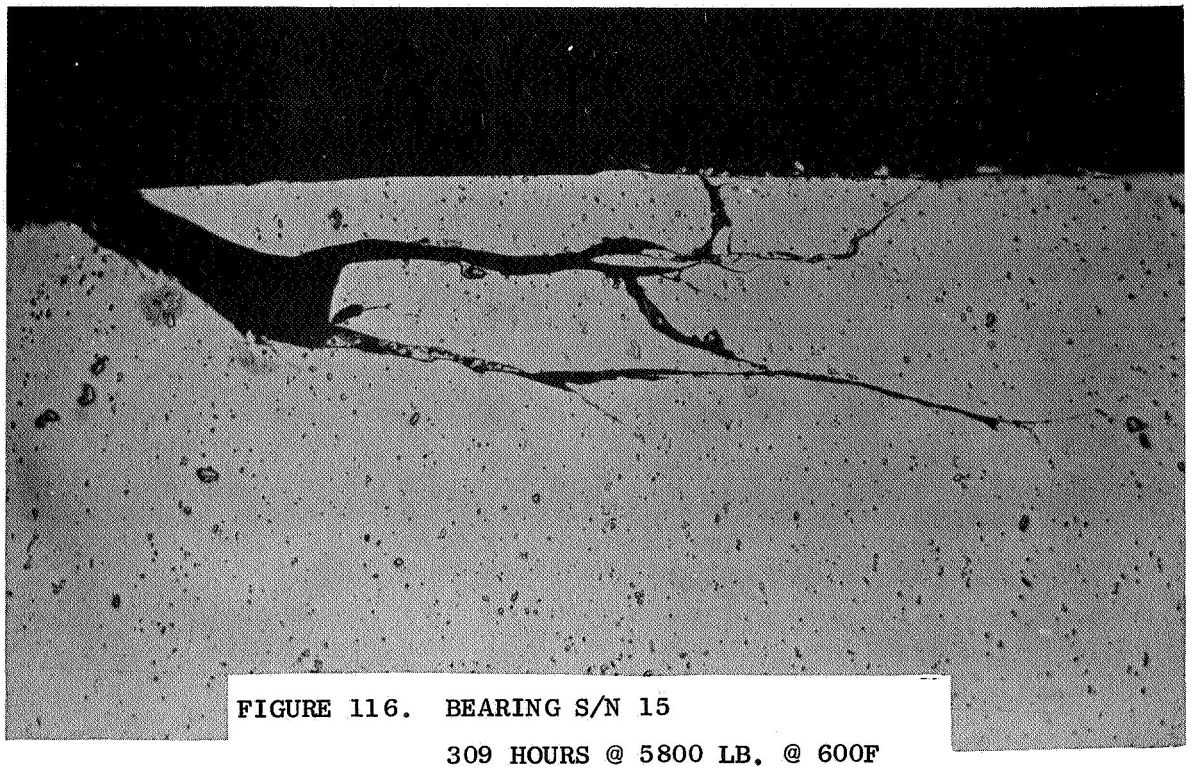


FIGURE 116. BEARING S/N 15

309 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM-177F
BALL FATIGUE FAILURE
MATERIAL: CVM-M-50
MAG: 250X **UNETCHED**

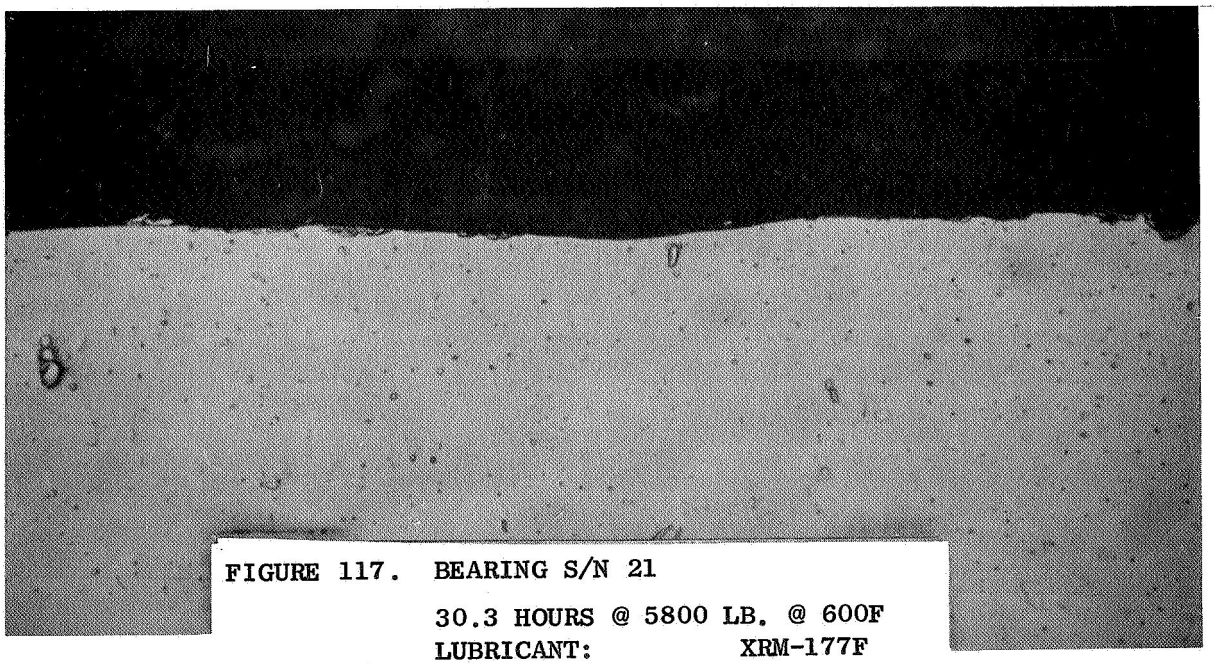


FIGURE 117. BEARING S/N 21

30.3 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM-177F
BALL WEAR
MATERIAL: CVM-M-50
MAG: 500X **UNETCHED**

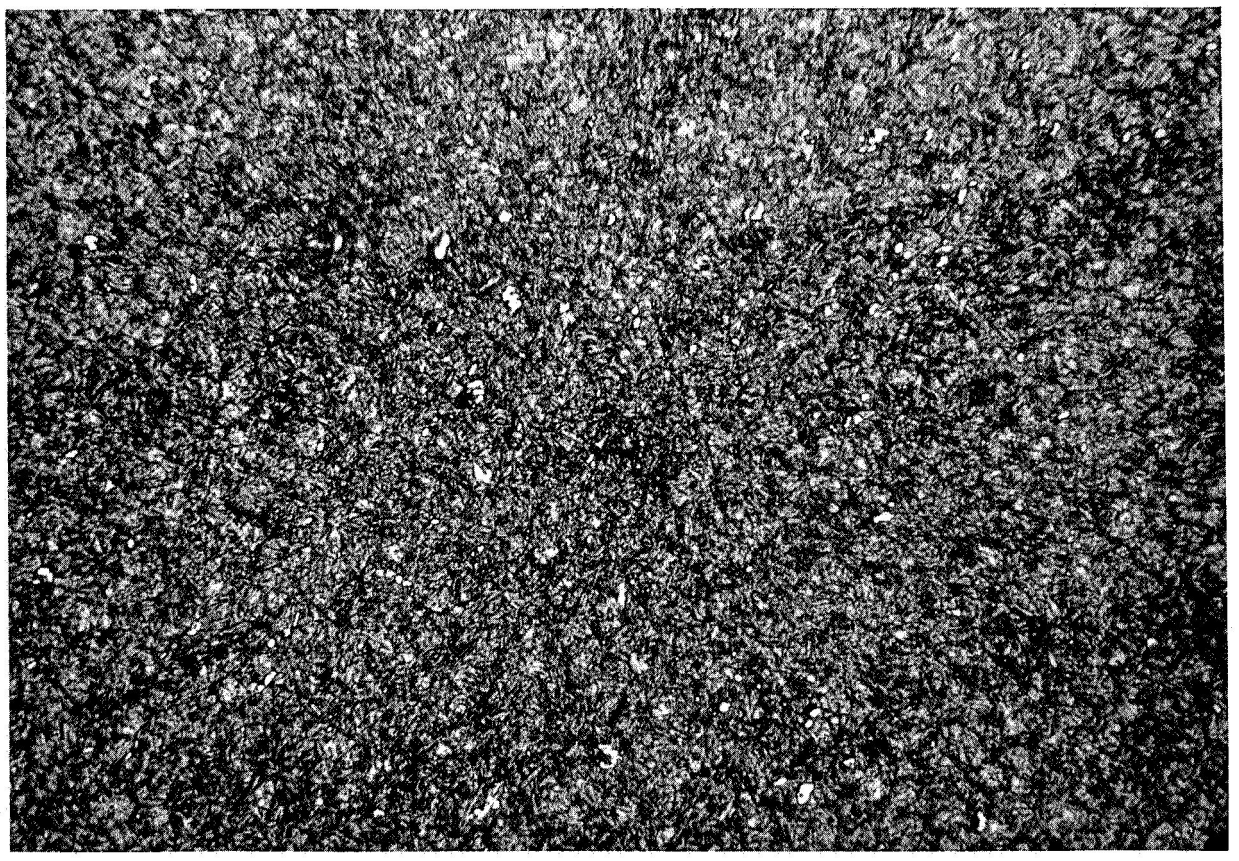


FIGURE 118. BEARING S/N 21

30.3 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM-177F
TYPICAL STRUCTURE: INNER RING
MATERIAL: CVM-M-50
MAG: 500X ETCHANT: 3% NITAL

FIGURE 119. BEARING S/N 21

30.3 HOURS @ 5800 LB. @ 600F
LUBRICANT: XRM–177F
SURFACE OF BALL, SHOWING WEAR
MATERIAL: **CVM-M-5**0
MAG: 1000X ETCHANT: **3%** NITAL

In visual examination of the fatigue failures encountered with the MCS-354 it was noted that the failures were somewhat different than those obtained with either of the other two fluids. Despite this, however, the appearance of the failures on a micro basis is very close to what might be called classical fatigue except that the nucleus of the failure is closer to the surface than would be expected based on calculations of the depth of maximum sub-surface .shear stress. This is illustrated in Figures 120 and 121.

The WB-49 material which exhibited a relatively poor fatigue life, showed a micro-structure which is not considered conducive to long rolling element fatigue life. A metallographic examination was conducted on the failed WB-49 bearings and typical photomicrographs are shown in Figures 122 through 124. Of primary interest is the massive carbide banding noted in the structure of this material. This is illustrated specifically in Figure 123 which shows the extremes of carbide banding noted in the structure. There is no question that this banding is one of the primary reasons for the short lives being encountered by these bearings, This type of structural deficiency does however, raise the question whether the material being tested is typical of the particular material. There is always the possibility that the specific material being evaluated was not produced with optimum forging practices.

However, in examining the structure of the RC Rig test bars, as well as the structures illustrated in a NASA report,[10] it appears that the massive carbide banding is common in this material. This is not too surprising in view of the high percentage of carbide forming elements present in this material. In view of this, it appears then that in order to derive the maximum usefulness from this material, improved techniques will be required in the reduction of the billet to the final shape. Obviously, as a first step, the severe carbide banding will need to be broken up and homogenized.

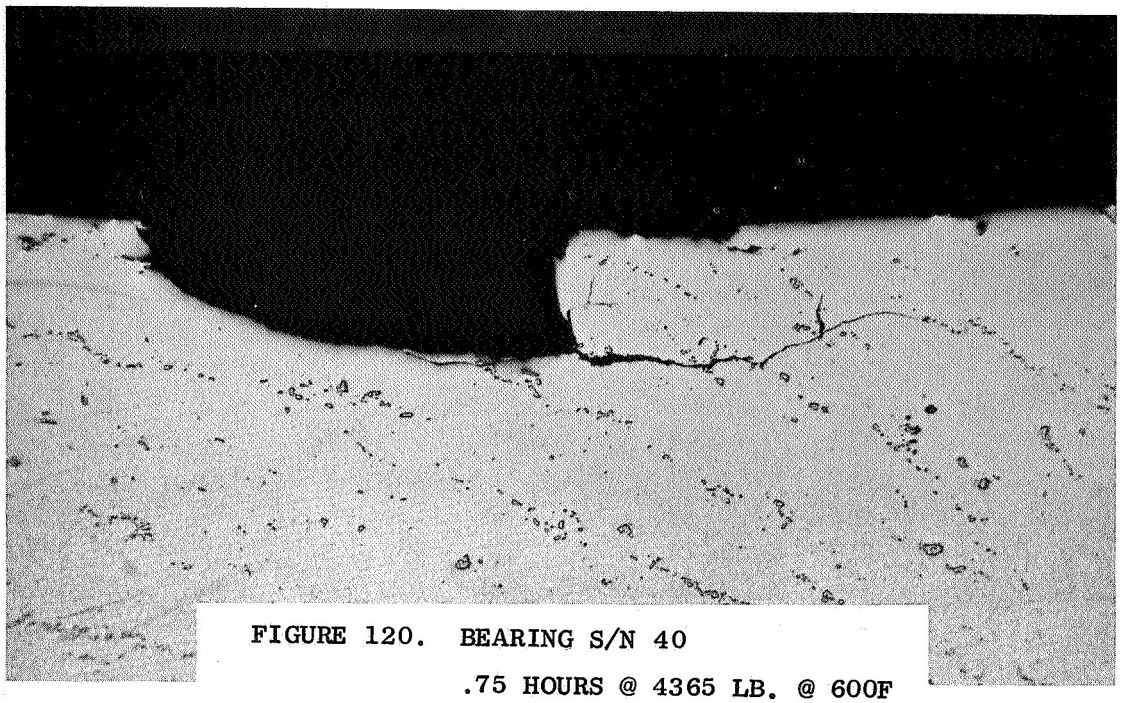


FIGURE 120. BEARING S/N 40

.75 HOURS @ 4365 LB. @ 600F
LUBRICANT: MCS-354
INNER RING FATIGUE FAILURE
MATERIAL: CVM-M-50
MAG: 200X UNETCHED

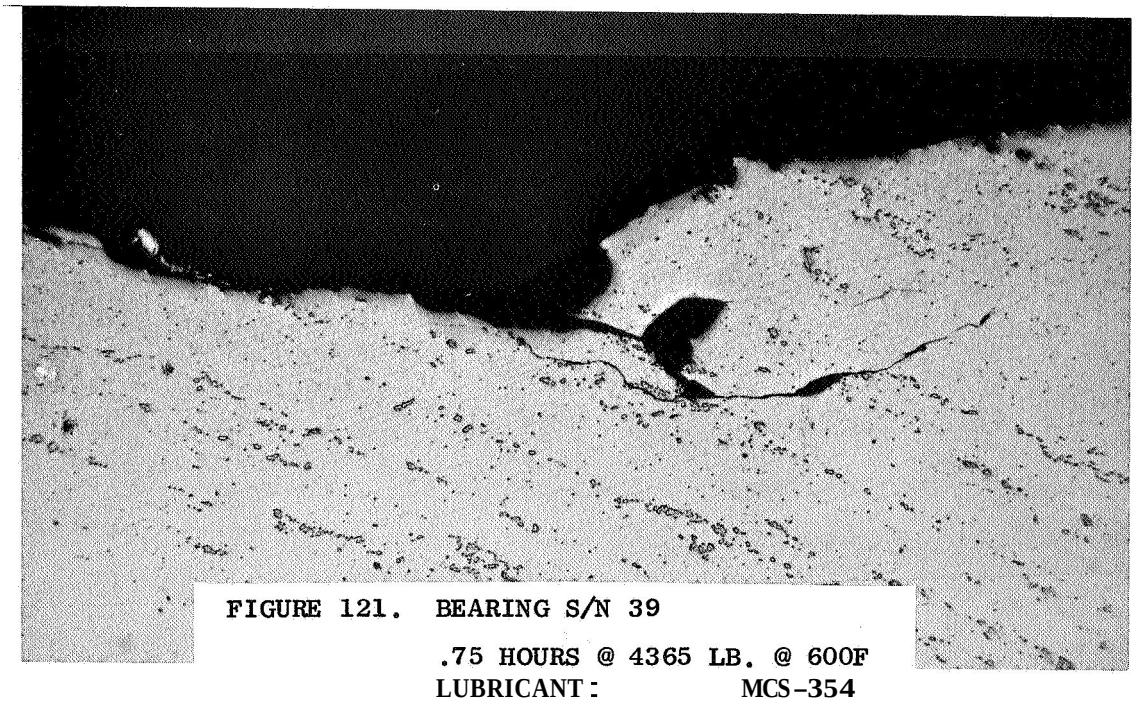


FIGURE 121. BEARING S/N 39

.75 HOURS @ 4365 LB. @ 600F
LUBRICANT: MCS-354
INNER RING FATIGUE FAILURE
MATERIAL: CVM-M-50
MAG: 200X UNETCHED

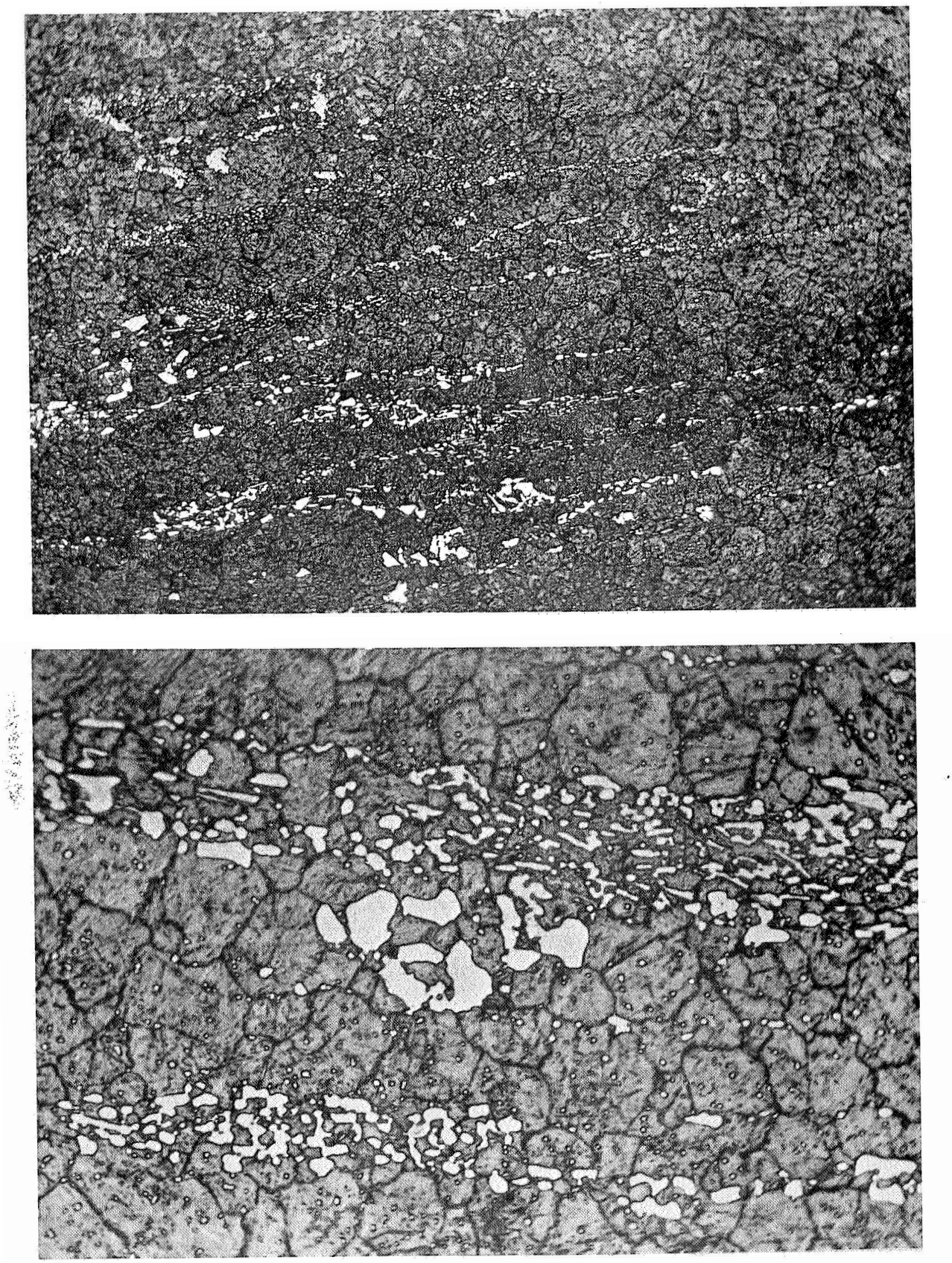


FIGURE 122. TYPICAL MICROSTRUCTURES OF WB-49.

BEARING S/N 256. NOTE MASSIVE CARBIDE BANDING.

MAG. TOP: 250X ETCHANT: 3% NITAL
BOTTOM: 1000X

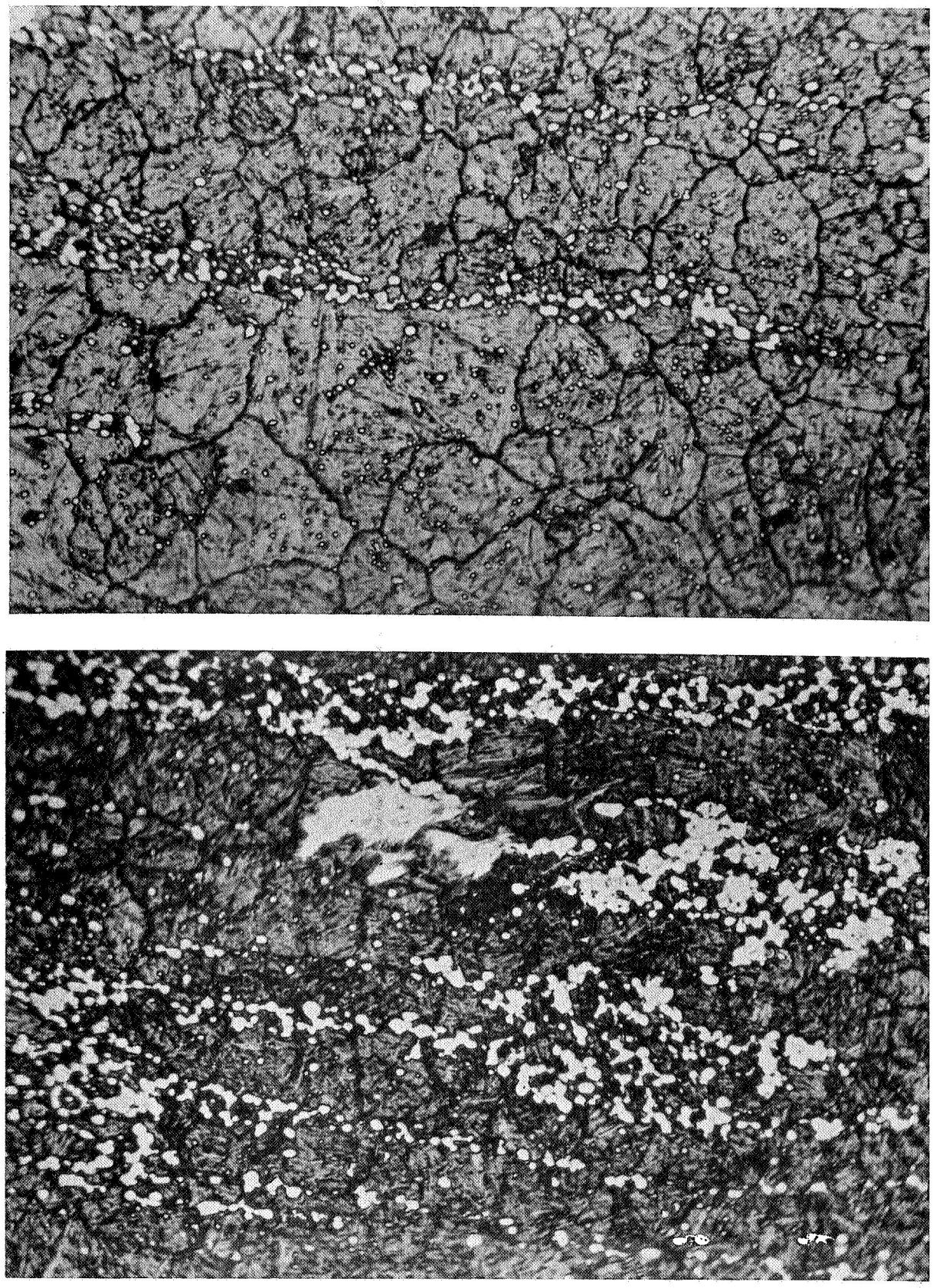


FIGURE **123.** TYPICAL EXAMPLES OF MASSIVE CARBIDE BANDING IN WB–49

TOP SHOWS MINIMUM BANDING, WHEREAS BOTTOM PHOTOGRAPH SHOWS AREA OF MAXIMUM BANDING.

MAG: 1000X ETCHANT: **3%** NITAL

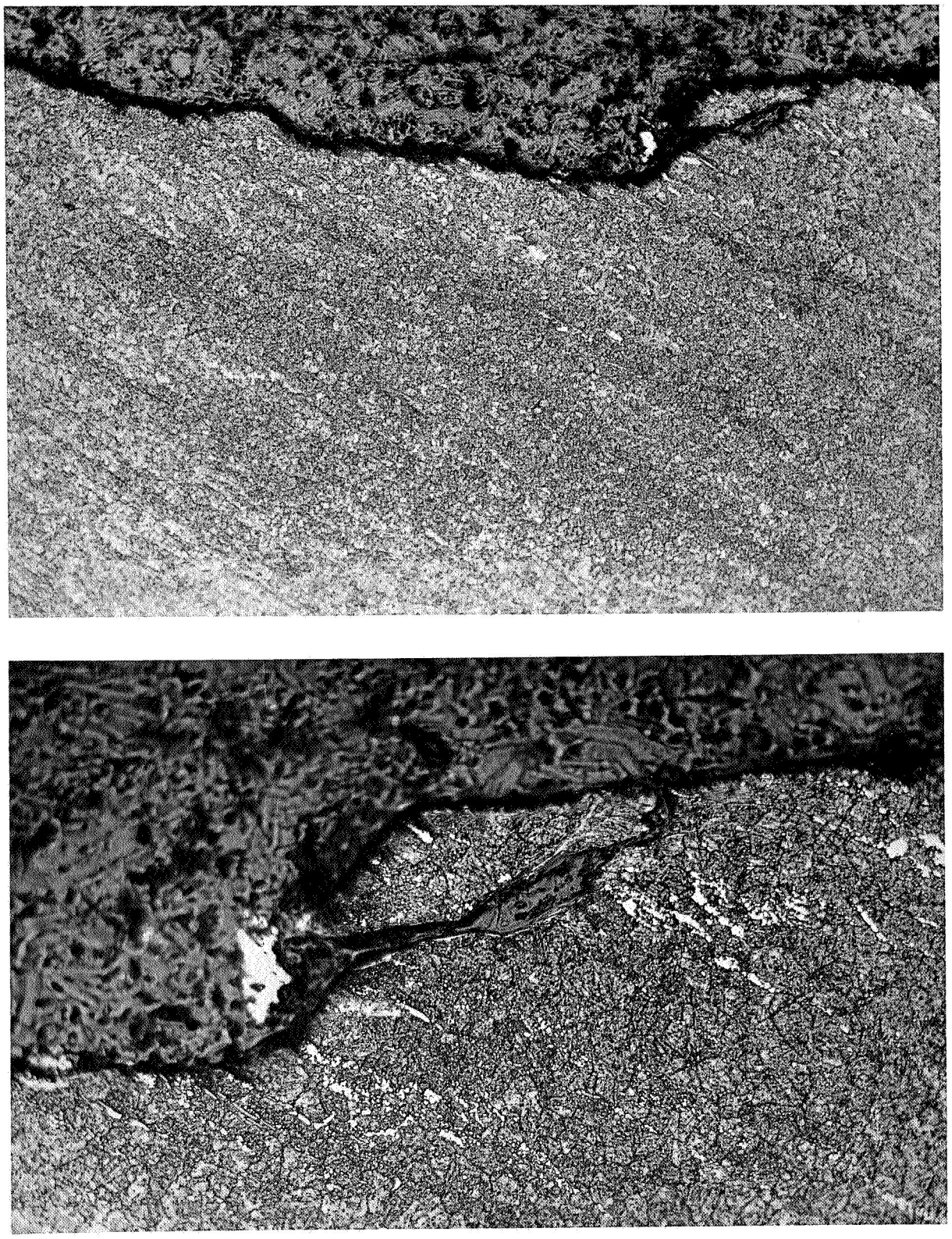


FIGURE 124. INNER RING FATIGUE FAILURE

BEARING S/N 256

TIME TO FAILURE: 26 HOURS
MAG. TOP: 100X ETCHANT: 3% NITAL
BOTTOM: 250X

8.0 Lubricant Analysis

A continuous check was maintained on the test lubricants during the time the lubricant was in the bearing testers, Samples of oil were taken at approximately 24 hour intervals from the individual testers and analyzed for viscosity, acid number and contamination.

Viscosity measurements were performed at 100F and per ASTM specification D445. Neutralization number measurements were made per ASTM specification D974-58T. KOH is used as the titrant, with para-naptholbenzein used for the indicator. Contamination was checked using infra-red spectrophotometric methods, This test was established primarily to assure that the test fluid had not been contaminated with another fluid. In the tester set-up, this could occur if seal leakage allowed the MIL-E-7808 slave bearing lubricant into the tester cavity. No evidence of this was ever noted indicating the efficiency of the seals being used,

The viscosity measurements made on the XRM-177F generally indieated an initial decrease in viscosity with time, This was expected as at the 600F temperature, some cracking of the oil must be expected. The slight decrease in neutralization number observed was not considered significant in view of the initial low value of the as-received fluid. Actually, the deviations of the samples, during and after test, from the initial ones are within the accuracy range of the test method. These measurements are additional evidence that this oil is capable of operating under the severe environmental conditions of this test,

While there was some discoloration observed in the oil after test, these visual observations also tend to indicate that no significant thermal breakdown of the oil is occurring at the test temperature and under total N_2 inertion. Figure 125 is a typical viscosity-time curve for the XRM-177F. This curve is representative of the measurements made on the 600F test series.

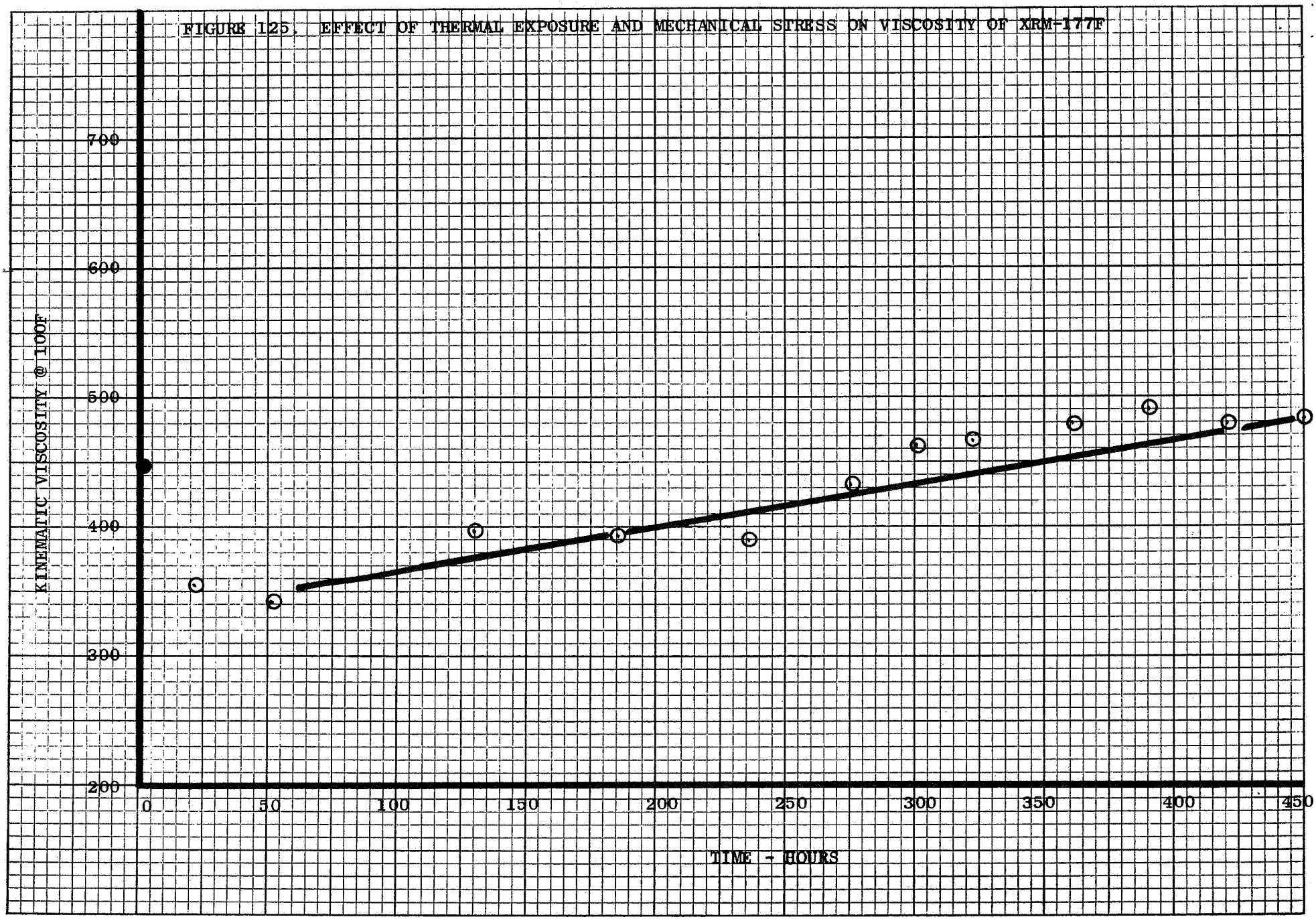
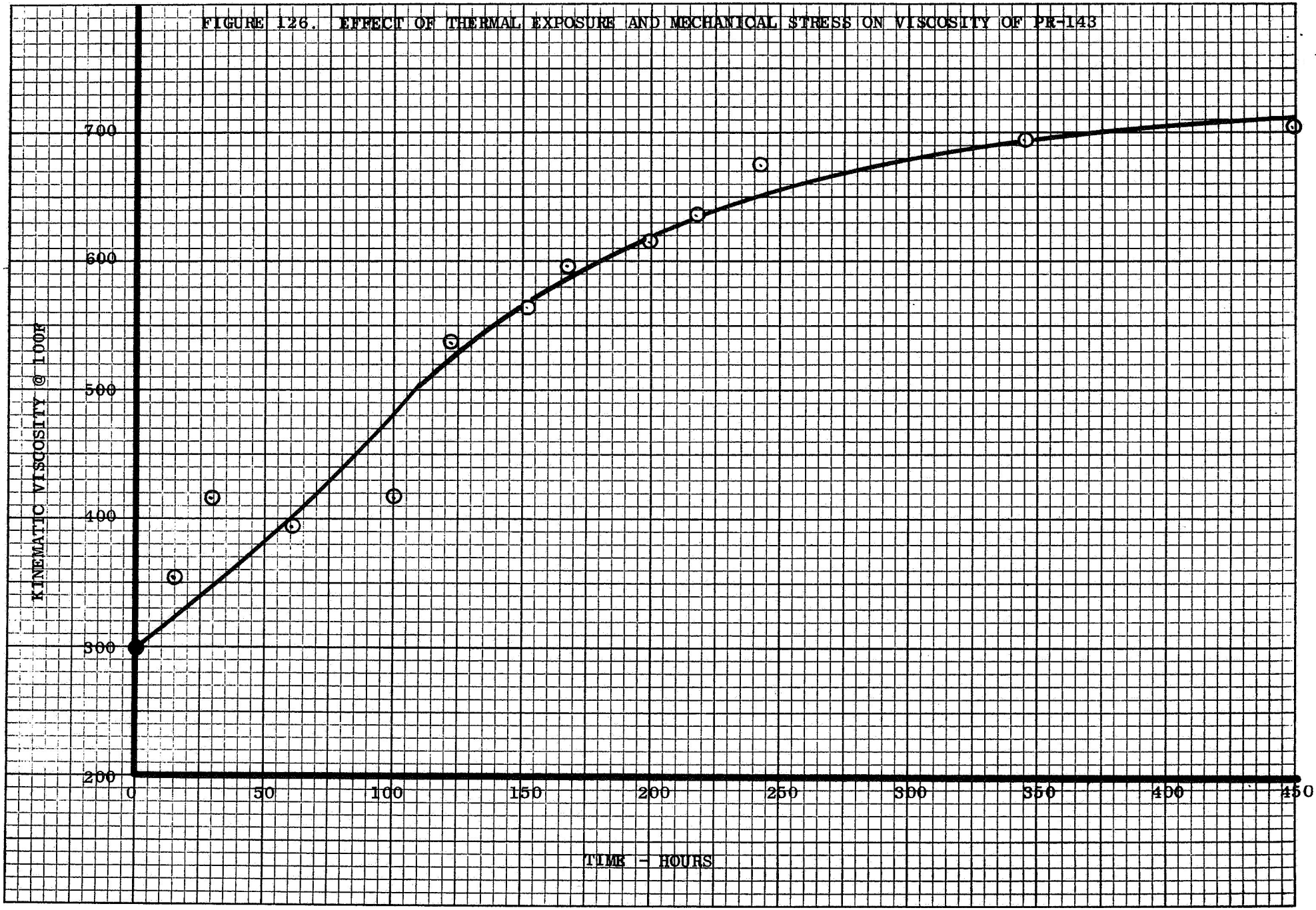


FIGURE 125. EFFECT OF THERMAL EXPOSURE AND MECHANICAL STRESS ON VISCOSITY OF XRM-177F
KINEMATIC VISCOSITY @ 100F
700
600
500
400
300
200
0
50
100
150
200
250
300
350
400
450
TIME - HOURS

With the MCS-354, a definitive trend is somewhat more difficult to determine, mainly because the high volatility of the fluid required relatively large quantities of "make-up" oil. Consequently, the periadic diluent effect of these oil additions made it difficult to establish the true viscosity changes. However, it does appear that the MCS-354 shows a slight overall, increase in viscosity with time. A representative viscosity-time curve:. is presented in Figure 126.

Lastly, the PR-143 shows a significant increase in viscosity after a relatively short period of time followed by a more gradual but constant increase as time progresses. This behavior is as expected with a polymeric fluid, where the light ends are volatilized leaving the heavier, high viscosity ends. This is illustrated in Figure 127 which shows representative viscosity-time relationships for the PR-143.

FIGURE 126. EFFECT OF THERMAL EXPOSURE AND MECHANICAL STRESS ON VISCOSITY OF PR-143



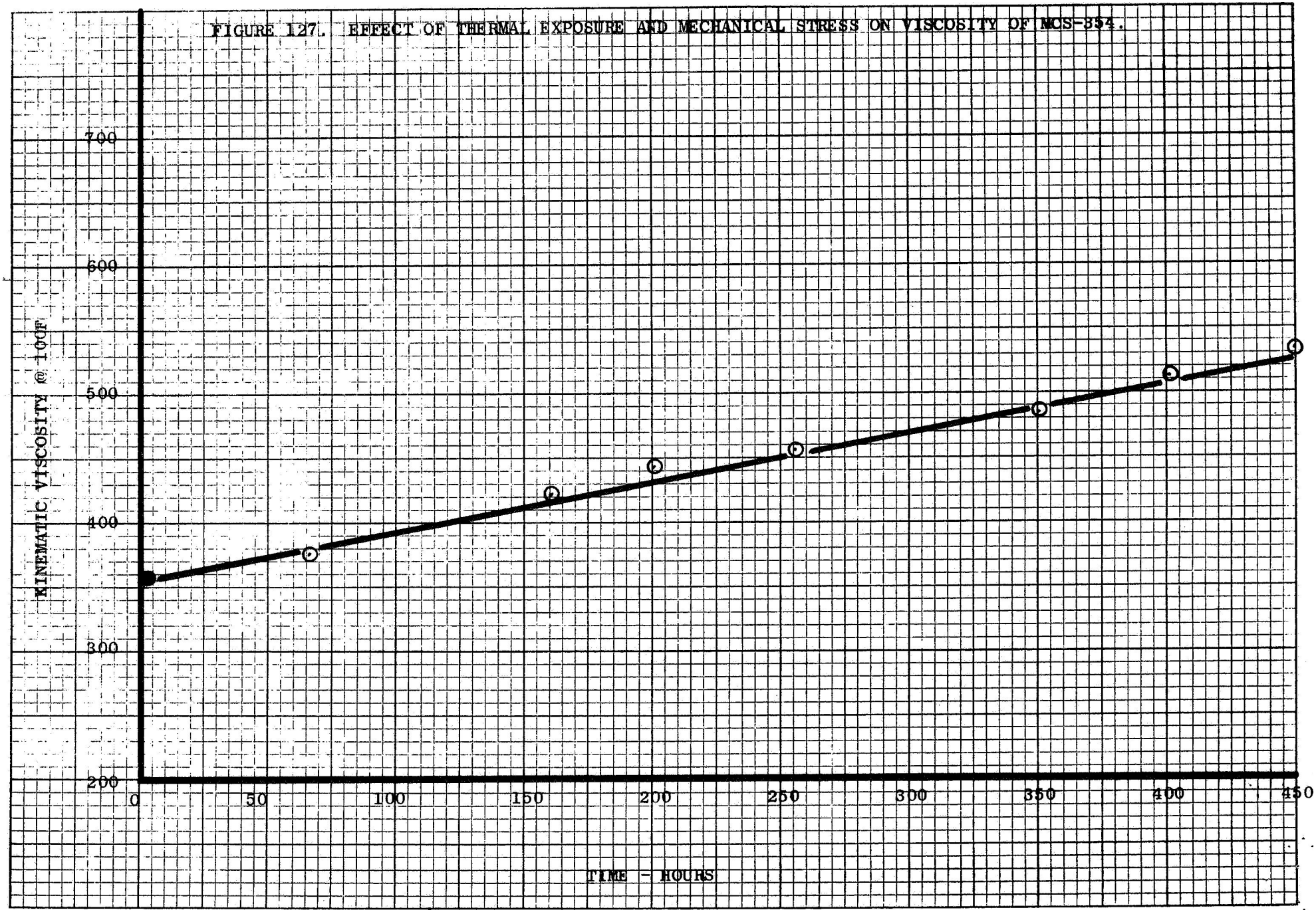


FIGURE 127. EFFECT OF THERMAL EXPOSURE AND MECHANICAL STRESS ON VISCOSITY OF MCS-354.

9.0 Bearing Measurements

To assist in the evaluation of the bearing data, a complete record was kept on all bearing and bearing component dimensions, and weights, both prior to and after testing. Additionally, surface profile measurements were made on the inner and outer bearing raceways prior to and after testing.

The individual bearing components which were weighed were the outer ring, the two segments of the split inner and the balls., These measurements were made on a scale calibrated into one-hundredths of a gram. Surface profile measurements were made on the outer and inner raceways using a Proficorder with measurements made at three locations around the tracks.

An analysis of all of these data does not add too much to the general body of knowledge regarding these bearings. The weight measurements indicate that, in the XRM-177F fluid there is essentially no measurable weight change in the bearing components. This, of course, is totally expected as no visual evidence of wear was observed in the bearings tested with this fluid, In the case of the MCS-354 there are some rather drastic weight losses, particularly in the ball components, although even here, there is considerable variation from one bearing to the other. In the bearing components tested with the PR-143, there is also a relatively random change in the weight measurements.

The surface profile measurements also indicate basically the same type of information. On those bearings tested with the XRM-177F, the surface profiles indicate essentially no deformation or wear, indicating that a total elastohydrodyaamic film was maintained throughout the course of testing, With the MCS-354 there is evidence of wear as well as some plastic deformation Caking place, although there is no discernible pattern to these effects. The surface profile traces are extremely difficult to interpret due to the poor condition of the raceway surfaces. The surfaces of the bearings lubricated with the fluorinated polymer, PR-143, indicate

generally very little wear and essentially no plastic deformation, again lending support to the theory that this lubricant does provide a mixed, if not total, elastohydrodynamic film at the 600F temperature. However, as was the case with the weight measurements taken on these bearings, there was some disagreement in the data in that some of the surface profile measurements indicated a total lack of wear, whereas others did indicate some evidence of abrasive metal removal due to asperity contacts. Some of this is also most likely due to the corrosion observed on the bearing components.

The major conclusion which can be drawn from these attempts at measurements are that either the large size and/or weight of the bearing per se, or the relatively heavy load placed on the bearing, put it beyond the realm of extremely precise measurement such as can be made on smaller, perhaps less lightly loaded bearings or on bench rig test specimens. This is particularly true in the case of the weight measurements where the total weight of the components is such that a small amount of wear resulting in a slight weight change would be extremely difficult to detect and pinpoint, Visual observations such as the condition of the ball track surfaces, or actual physical measurements such as were used to determine the changes in ball dimensions are perhaps a much better and certainly more economical means of determining what type of lubrication regime existed at the test conditions.

10.0 Discussion

The work reviewed in this report has provided a good insight into the feasibility of operating bearings at temperatures that exceed today's conditions by at least 150° to 200°F. The demonstration that conventionally designed bearings can operate successfully at 600F is an important milestone in the state-of-the-art of bearing technology, and should serve as the basis for additional high temperature bearing and lubricant development and evaluation. It will also serve as a guideline for the future when it is expected that the test temperatures employed in this program will be commonly seen in high performance turbo-jet engines.

Before proceeding into a discussion of the results, it might be well to briefly review the rationale' of conducting a test program such as the one discussed in this report under conditions which are not at the present time contemplated for actual engine operation. Specifically, this refers to the use of the nitrogen atmosphere used in the current series of tests. The total inertion of a propulsion system is an extremely difficult endeavor and would be considered only when all other means to accomplish the same purpose had been exhausted. Consequently, it might be asked why a test program was conducted under such currently unrealistic conditions. The reason is actually quite simple, insofar the prime purpose of the program was to establish the capability of three advanced fluids to sustain elastohydrodynamic bearing operation under extreme temperature conditions. Therefore, as is the case with many basic engineering studies, a test environment had to be selected so that the primary purpose of the test program could be achieved. The data obtained is valid whether it be conducted in a nitrogen system or in a more normal air environment, It is the basic data which is of interest and the generation of which justifies the use of a controlled environment.

A parallel to this can be drawn in much of the high temperature metallurgical testing which is conducted. Much of this testing is conducted in either an inerted atmosphere or in a vacuum. Obviously, the materials

being evaluated will generally not operate in these environments. In order to obtain the data, however, it is necessary to control the atmosphere so that proper, reliable, and repeatable engineering information can be obtained, It is expected that under the impetus of the increasing performance requirements made on jet engines, the lubricant manufacturers will develop fluids which will be viable at temperatures approaching 600F and which will at the same time, be thermally stable to the point that they can operate in an air atmosphere. When these fluids are available, the information obtained in this study will be directly applicable and will serve as a guideline to judging the efficacy of the lubricants.

The three fluids evaluated in this program are representative of the most advanced high temperature materials available today. They also represent three distinctly separate families of fluids, and as such, it is logical to assume that the data obtained will be representative of other fluids of the same generic base stock. Based on the test results, the synthetic paraffinic oil, XRM-177F, is presently the most promising candidate for high temperature bearing lubrication. The appearance of the bearing components and particularly, the rolling elements and raceways, gave full evidence that elastohydrodynamic lubrication was maintained in the 600F temperature range with this fluid. Obviously, the same comment applies to the 500F and 400F temperature range. The best evidence of this was the appearance of the balls and raceways which, when observed at high magnification, still showed the original grinding marks, indicating that no asperity contact had taken place. Secondly, the surface profile measurements made after testing also indicated that essentially no changes had occurred in the stressed areas of these bearings. As discussed previously, the surface profile measurements were in general not highly conclusive, except in the case of the XRM-177F where no indication of wear and/or plastic deformation was observed. Lastly, the weight measurements made on the bearings prior to and after the test which, while also not conclusive, did indicate that with the XRM-177F, no weight changes had taken place,

further indicating the absence of wear. Highly encouraging also were the relatively small changes in lubricant viscosity and acid number observed after exposure of the fluid to a prolonged period of time at high temperature, This is quite significant, as it tends to indicate a long time thermal stability for this fluid. The data obtained by General Electric on this fluid also agrees well with that obtained by other organizations engaged in performing similar testing.(1)

Lastly, the data on the XRM-177F clearly indicates that this fluid is useful in the total temperature range of interest, i.e., 400F to 600F. This is not unexpected considering the good viscosity index of this fluid.

Compared to the XRM-177F, performance of the modified polyphenyl ether, MCS-354 was not satisfactory. The inability of this fluid to operate in an inerted atmosphere was not too surprising, as there are a number of fluids which require oxygen to form reactive boundary lubricants. However, the subsequent continued poor performance in an air atmosphere was disappointing, The operation of the bearings at 600F with the MCS-354 was characterized by extreme wear which necessitated a shut-down of many tests prior to the initiation of any normal fatigue failures. There was definite visual evidence of the lack of elastohydrodynamic conditions. The rolling elements as well as the races exhibited evidence of burnishing and polishing with most of the original grinding marks being completely obliterated, indicative of mixed or boundary lubrication. In most instances, surface distress was severe enough so as not to allow a meaningful surface profile analysis. Also, the weight measurements, while not consistent, did indicate significant weight losses, particularly on the balls and to a lesser extent, on the inner and outer rings. In the few instances where the bearings failed in fatigue, visual examination showed these failures to be more apparently surface initiated rather than the more classic sub-surface originating spalling. This was a deceptive observation, however, as metallographic examination indicated that the failures did initiate sub-surface at about the depth of the calculated maximum shear stress region.

The most interesting fluid of the three investigated was the fluorinated polymer, PR-143. Based on the chemical make-up of the three fluids, the polymeric material has the greatest potential as an extremely high temperature lubricant. It exhibits a number of properties which make it extremely attractive to engine designers, chief among which is its inherent fire safe operation. Conversely, the corrosive aspect would tend to make it difficult to apply in existing lubricating systems, although this should not be a major problem in advanced engines, where the lubricating system can be designed with materials which will accommodate the potential corrosive effects of this material. On this subject it should be pointed out that the fluid did not exhibit the severe corrosive effects which had been anticipated and in fact, the overall damage to either the test facilities or the test bearings themselves was within tolerable limits.

While the PR-143 gave fatigue lives close to those predicted by RECAP analysis, the performance of the fluid in the test series was not consistent. Unlike the XRM-177F where obvious elastohydrodynamic lubrication conditions existed at 600F and unlike the polyphenyl ethers, where just as obviously boundary conditions existed, there appears to be evidence of both conditions in the PR-143 between various tests. As can be seen from Table 14, there were tests which resulted in fatigue failures with none or a minimal amount of wear. There were also tests which were aborted because of rather extensive wear, and there were some tests which were terminated at 500 hours, which exhibited extremely good surfaces with *no* evidence of incipient fatigue failures or significant amounts of wear. One can only assume that this fluid is perhaps more sensitive to minute variations in bearing geometry or very minor changes in the test conditions.

The most serious drawback of the fluorinated polymer appears to be its tendency to increase the difficulty of bearing heat rejection, most likely due to its high density as well as its low thermal conductivity. It was

observed that the PR-143 test bearings when operated under identical conditions as the other two fluids, tended to stabilize out at a temperature approximately lOOF to 125F above that observed on the other bearings, Coincidentally, it was noted that the horsepower requirements required to drive the testers was increased significantly and in fact resulted in a rash of drive belt failures. It could be assumed that the primary cause of this, is that in such a situation the retainer will act as a water brake generating considerable heat in the process. This increased bearing stabilization temperature with the PR-143 has also been noted by other organizations evaluating this fluid in high temperature bearing operations. At the present time, this factor becomes an extremely serious one in terms of applying this fluid into an engine. There is no question that, if this material is to be used, it will be necessary to re-evaluate the entire cooling system of the engine, or to make drastic changes in the basic bearing design to accommodate the fluid. Despite this, however, it is believed that this fluid has considerable promise, especially in view of its performance demonstrated in this program. This performance was achieved without the benefit of any type of additive packages such as EP or wear improvers.

In the analysis of the data there are a number of interesting observations which can be made, although the most intriguing and perhaps the most significant from a standpoint of further design of high-temperature lubricants, is the apparent viscosity/life relationship which can be demonstrated at the 600F level. Figure 128 shows the viscosity temperature relationships of the three test fluids, It can be seen that the XRM-177F and the fluorinated polymer are relatively close and have essentially the same viscosity index. The polyphenyl ether on the other hand, has a less attractive viscosity index and deviates somewhat from the other two fluids. If one now establishes a simple relationship such as plotting the log of the B_{10} life versus the kinematic viscosity, the curve shown in Figure 129 can be constructed. While it is conceded that

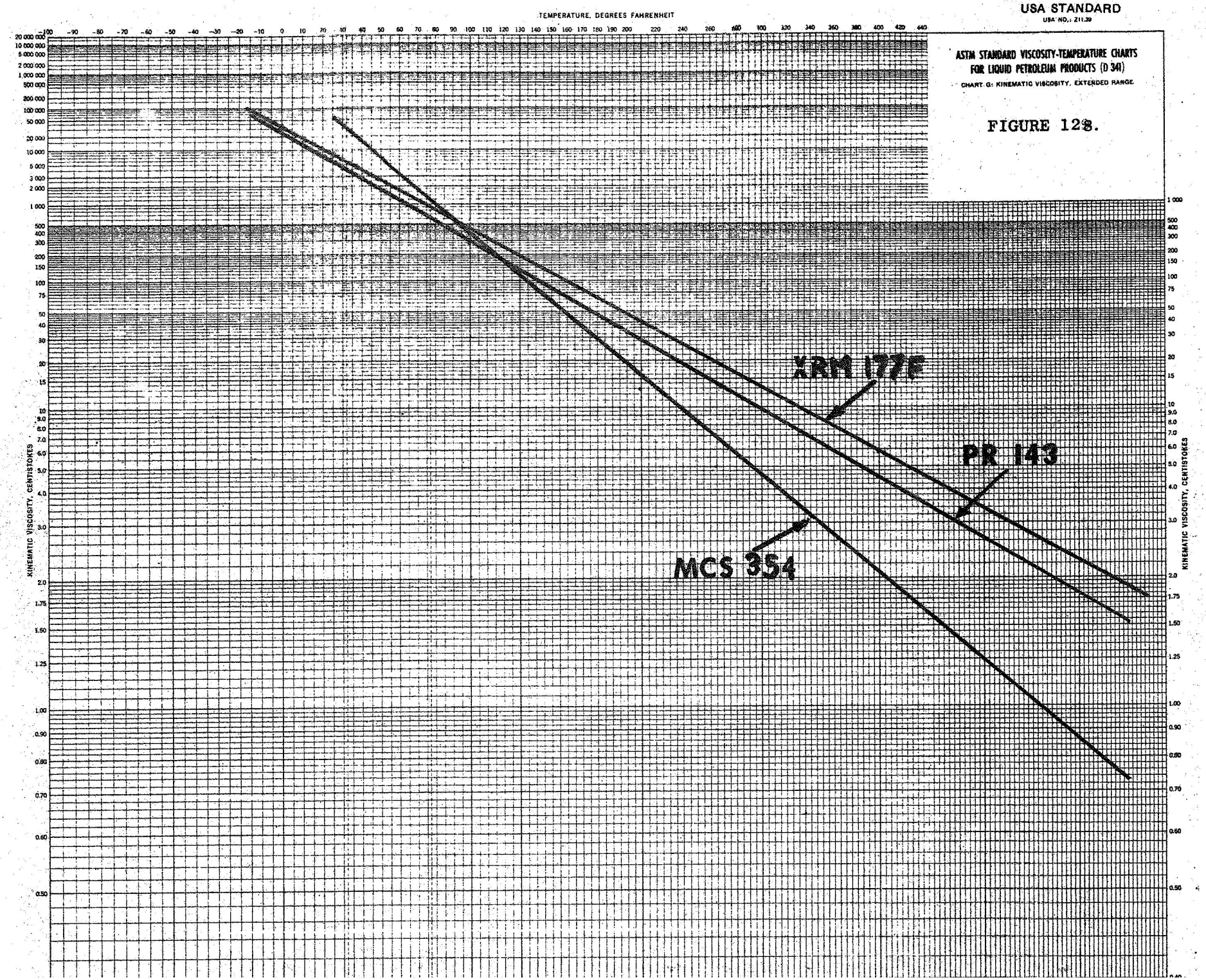
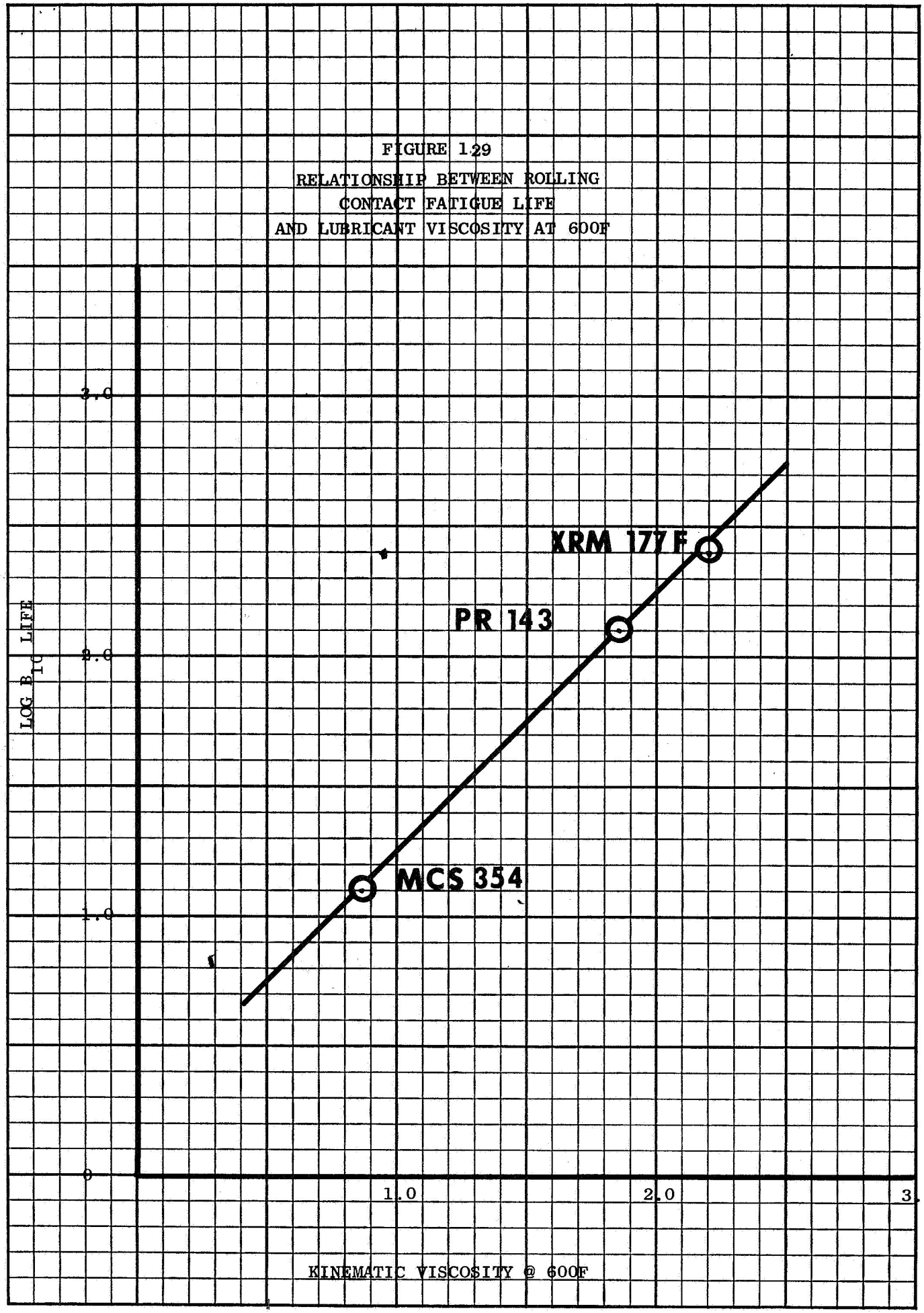

USA STANDARD
USA NO.: Z11.39
ASTM STANDARD VISCOSITY-TEMPERATURE CHARTS
FOR LIQUID PETROLEUM PRODUCTS (D 341)
CHART G: KINEMATIC VISCOSITY, EXTENDED RANGE
FIGURE 128.
TEMPERATURE, DEGREES FAHRENHEIT
KINEMATIC VISCOSITY, CENTISTOKES
XRM 177F
PR 143
MCS 354

FIGURE 1.29
RELATIONSHIP BETWEEN ROLLING
CONTACT FATIGUE LIFE
AND LUBRICANT VISCOSITY AT 600F



in view of the scarcity of data for the polyphenyl ether, the B_{10} life is at best a poor estimate, the straight line relationship which is achieved is quite remarkable. Pursuing this argument somewhat further, if one assumes that the B_{10} lives indicated on the subject curve are indicative of bearing tests where true elastohydrodynamic lubrication existed, even in the case of the polyphenyl ether, then one might speculate that, in order to achieve bearing life on the order of AFBMA or better, the lubricant viscosity must exceed 1.5 CS at the operating temperature. If this is indeed the case, then a useful guideline has been established for the development of fluids designed to operate at extreme temperature levels.

The extremely disappointing performance of the WB-49 bearing material is thought to be a result of the basic difficulty of working this material to achieve a sound, homogenous structure. This has been pointed out previously in that the severe carbide banding which was observed unquestionably, contributed to the very early failures observed with this material. However, since reviewing this condition with other sources which have evaluated this material, it can only be assumed that at the present time, this is the best structure which can be obtained with this material and if this is indeed the case, the material cannot be considered for extended reliable high temperature bearing applications.

While on the subject of bearing materials, two other significant items were identified as a result of this program. The first of these is the excellent performance of the M-50 at the 600F temperature. This certainly provides a high degree of assurance that M-50 as presently constituted will be useful for quite some time to come, and that at the present time the limitation on high temperature bearing operation is still the lubricating fluid. Secondly, the performance of the S-Monel cage material was highly gratifying. While there were a few isolated cases of cage failure, in general, the performance of this part was extremely reliable. In most cases the cage pocket exhibited a very minimal amount of wear and no deformation. This can be seen from the several photographs which have been used

to illustrate the condition of these parts. On the other hand, the performance of the tool steel (M-1) cages used in conjunction with the WB-49 bearings was somewhat less reliable. Approximately 2% of the tool steel cages exhibited fracture failures. Consequently, based on these data it appears that the S-Monel may be a more attractive material for high temperature bearing applications.

Lastly, the results of the RC Rig tests, when viewed against the full-scale bearing test data, again illustrates the usefulness of such bench-type testing in predicting the behavior of large bearing test results. This is illustrated in the tabulation shown below:

RC Rig/120 mm Bearing Test Results @ 600F

		B_{10} Life				Ranking	
Brg. Matl.	Fluid	AFBMA RC Rig* (Stress Cycles)	Actual RC Rig (Stress Cycles)	RECAP Bearing* (Hrs.)	Actual Bearing (Hrs.)	RC Rig	Brg.
M-50	XRM-177F	1.4×10^6	2.35×10^6	100	252	1	1
M-50	MCS-354	1.4×10^6	0.69×10^6	100	12.7	3	3
M-50	PR-143	1.4×10^6	1.04×10^6	100	128	2	2
WB-49	XRM-177F	1.4×10^6	1.7×10^6	100	84		

* Corrected for CVM M-50

It can be seen from the above data that the RC Rig ranked the three fluids in exactly the same order as the large scale bearing test and in fact, the percentage of difference between the fluids in relationship to the AFBMA rating is in very good agreement, The RC Rig also predicted a lower fatigue life for WB-49, as compared to M-50, although here the magnitude of failure severity was much greater in the actual bearings than had been predicted with the RC Rig. Again, this is thought to be due to the fact that the RC Rig bars did achieve perhaps a more uniform working and consequently, contained a more homogeneous structure than the large bearing components.

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Wilmington, Delaware 19899

Airesearch Manufacturing Company
Department 93-3
9851 Sepulveda Boulevard
Los Angeles, California 90009
Attention: Hans J. Poulsen (1)

General Electric Company
Silicone Products Department
Waterford, New York 12188
Attention: J. C. Frewin (1)

Stewart-Warner Corporation
1826 Divcrsey Parkway
Chicago, Illinois 60614
Attention: R. F. Wharton (1)

Department of the Army
U.S. Army Aviation Material Labs,
Fort Eustis, Virginia 23604
Attention: J, W. White, Propulsion Division (1)

SKF Industries, Inc.
Engineering & Research Center
1100 First Avenue
King of Prussia, Pennsylvania 19104
Attention: L. B. Sibley (1)
T, Tallian (1)

Nyatt Division
New Departure Bearing Company
Sandusky, Ohio 44870
Attention: D. Ruley (1)

Timkin Roller Bearing Company
Canton, Ohio 44701
Attention: Dr. W. Littmann (1)

Caterpillar Tractor
Peoria, Illinois 61601
Attention: B. Kelley (1)

General Motors Research Lab.
Warren, Michigan
Attention: Nils L. Muench (1)

Department of the Navy
Naval Ship Research
Development Center
Anapolis, Maryland 21402
Attention: Paul Schatzberg (1)